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THE DESIGN AND MAKING OF SAILS.*

BY ADRIAN WILSON.

The first consideration in the matter before us is the making of a proper distribution of the areas of the drawing power of sails. In a great measure the making of a plan will depend, of course, on the form of the hull to which we wish to apply it. Boats of radically different lines will want different plans, and, to determine as to what is the correct thing for a given set of lines, we shall be obliged to resort to a comparison to be obtained

decided on a certain amount of area, it should be so drawn in on the plan as to bring the center of effort a little in advance of the lateral plane. The distance to be placed between these two points is to be determined, to a great extent, on the form of the boat, as to her inclination to shift these underwater centers as she increases her angle of heel. In the later designs, those built to our latest racing rules, we find a tendency in all these modern



A SHIP WITH DOUBLE TOPSAILS.

(Photo by W. H. Rau.)

from other boats which may have the same essential characteristics in design as the one we are to supply with a plan. Given one or two known quantities as a basis for starting the plan will simplify our work in its early stages. The points mentioned are to know the location of the various centers; that is, the center of buoyancy, and also one other most important point, the center of lateral plane.

Using the center of lateral plane as a starting point, and, having

boats to upset calculations which had become almost exact in our older designs.

The new boats with extreme overhangs, so designed as to increase their length of water-line as they increase the angle of heel, unquestionably shift the under-body centers, and the result is that while a boat may balance perfectly in a light wind or at a short angle of heel, she may be inclined to do something entirely different when held at an increased angle, and the question arises in my mind if anything can be done by which this can be obviated

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in a sail plan. The shape of a sail for a boat whose design contains elements so likely to cause the difficulty mentioned is a hard problem to solve.

Our method of finding the center of effort is a very simple one, and I find that it will work out correctly every time. But very seldom are we asked to design a sail plan for a new boat, as to-day this is all done for us by the designer; often, however, we are



SAIL MAKING BY MACHINERY.

(Photo by G. E. Dow.)

asked to correct the sail plan of some boat which is radically wrong and cannot be made to steer as she should, and we are often able to make a decided improvement in boats that were very much out. As an instance: We had one customer who insisted that we give him the benefit of our experience on his boat, one of our modern 25-foot L. W. L. boats. The boat carried a bad lee helm, and I found by questioning him that in a good whole-sail breeze she came nearly to a balance with a full mainsail and the smallest or No. 3 jib. We went over her plan very carefully, and after measuring her under-water profile, or sheer plan, we made up our minds that a mistake must have been made in the designer's office in locating her center of lateral plane. Of course, knowing that she nearly balanced under the sails as mentioned above, we saw that the center of effort in these two sails should be the center of effort for her mainsail and large jib, and, in order to bring this into its proper relation to the center of lateral plane, it became necessary to alter the whole rig. We shifted the mast aft and cut off the bowsprit, and changed the mainsail (as shown in Fig. 1), but had to still keep to the class area.

Now, with our center of effort seven inches forward of the center of lateral plane, the boat has worked in a most satisfactory manner and is all the owner can wish for; in fact, she has won a number of races from boats that easily beat her before the alterations.

And again, before going into the discussion of shape and proportion of sails, one other consideration must be given, as to the different conditions under which a boat may be used. By conditions under which she is sailed I mean weather and locality, and in using the word conditions I wish it understood as referring to the strength of the winds. In our experience as sailmakers almost the first question we ask a customer is: "Where do you use this boat?" We have found that it makes a vast difference as to the location, for at the different points along our coast we find the weather conditions so entirely different as to cause us to make very different recommendations for a sail made for use in Buzzards Bay or Long Island Sound or at Marblehead. In our opinion the sail plan for a 21-footer for use at Marblehead should be quite different from one for use in Buzzards Bay. At Marblehead the weather conditions are such that we would recommend a very high, narrow plan, while at Buzzards Bay we would recommend a lower, broader plan for the same design. The conditions at Marblehead are, on an average, light winds. Also, the wind currents are high up from the water, and what might well be

called streaky currents, the wind being found in veins, a peculiar feature of this locality, especially in light winds.

I have no real authority for this statement, but should judge that my statement would stand some test. At Buzzards Bay the prevailing winds are southwest, and strong and heavy, the sea breeze blowing in damp and strong. As we ship our sails to many different localities we find it quite to our advantage to inquire into and study these conditions. Possibly San Francisco Bay may be an interesting example of these conditions. At first it was a difficult problem with us, and, in fact, it took about two seasons for us to accomplish the best results there. The conditions there are peculiar in that the trade wind blows directly into the harbor and generally right up the bay, taking the channel or center of the bay. Starting out from the city side of the bay a yacht will have a soft, light wind blowing from four to seven miles an hour. As she approaches the channel or center of the bay she enters the strong current of the trade wind until, reaching its center, she has a wind of from 25 to 30 miles an hour, and goes out of this as she reaches the other side of the bay. So, practically, she sails from a light wind through a strong one into another light wind, and here we find a double condition to contend with. Sails made for use in these waters are really a compromise between what we would make for the two conditions alluded to earlier. Knowing the conditions of weather under which the boat would sail, it would be a choice of either one of the plans mentioned—that is, the high, narrow plan, the low, broad plan, or a compromise between the two.

Let us consider first the important characteristics of the light-weather sail plan. In all of these plans the angle of the gaff or peak of the sail is a most important factor and is deserving of your greatest consideration. We have found from experience that the high, narrow sail will not bear so high an angle of peak as the low, broad sail plan. In order to get the best results for light conditions, it of course becomes necessary as the hoist is increased to shorten the gaff, and take 65 degrees maximum and 50 degrees minimum as a basis for all peak angles. As I say, taking 65 degrees as a maximum, as the hoist is increased and the gaff is shortened, the angle of peak should be lowered. I am speaking of a restricted area.



SAIL MAKING BY HAND.

(Photo by G. E. Dow.)

In the present raceabout class an angle of 60 degrees on the following dimensions of mainsail will be found to give the best results: Hoist 22 feet 3 inches, head 13 feet 9 inches, foot 24 feet, and leach 37 feet; 480 square feet in the mainsail and 120 square feet in the jib. The illustration before you is what we consider one of our best 21-footer sail plans for light conditions. In the proportioning of one sail to the other, or in dividing the area into the two sails, mainsail and jib, this plan is a very excellent one and shows the result of careful study as to the best result to be obtained in dividing the 600 square foot area. (Fig. 2.)

To make a comparison of one or two plans I will take first one of our champion 25-footers, and illustrate the point as to over-peaking the mainsail by explaining how at first this was a very unsatisfactory sail, and, by a slight alteration, was so greatly improved as to make the boat an easy championship winner. The sail as seen by myself, under almost ideal conditions, was far from satisfactory, and had a very serious fault in that it showed entirely too much bag in the upper forward angle of the sail. If the peak was set up to the angle at which it was designed, the sail

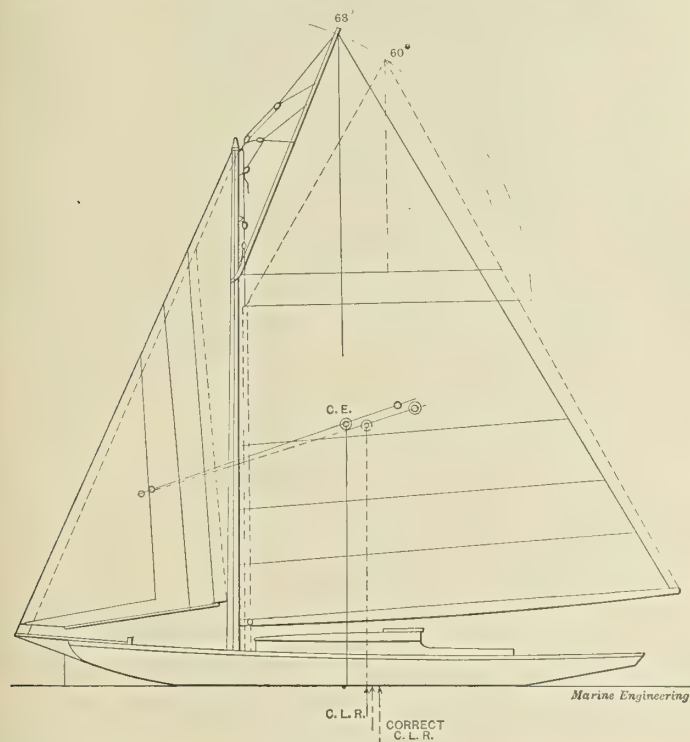


FIG. 1.

had the bad fault mentioned, and the yacht had to be kept off to allow the sail to be kept full of wind. Now, in order to lessen the bag mentioned, the peak was dropped, with the result that the gaff swung off to leeward, and then the boat would not reach as fast as she should. The correction was made by cutting down the angle of peak five degrees, with the result of getting an almost ideal sail. (Fig. 3.)

With the amount of area divided into the two sails of a jib and mainsail rig, care should be taken to so place the sails on the boat that one will be entirely independent of the other; that is, it will be found of much advantage if the jib is placed far enough forward so there can be no draught of wind or air current passing from the head-sail to the lee side of the mainsail. If the two sails are so close to each other that the current of wind from the jib strikes the lee side of the mainsail, it is more than likely to lift the forward part or luff of the mainsail out to windward, and, of course, that part so affected is a dead loss to the boat. The sails should be so placed that every inch of the mainsail may be doing its full work. Especially should the head of the jib be kept well away from the mast, as the upper part of the head of the jib is much nearer the mast than it is at the deck. Also, the upper part of the mainsail is at a broader angle than the boom. So you may readily see the importance of keeping the jib clear of the mainsail.

Possibly I should have given you earlier the names of the different sides of the sail. On a mainsail that part which is next to the mast is called the luff; that part following the line of the boom is the foot; and that part following the gaff is the head. The after side of all sails is the leach. The same applies to the jib. There the luff is that part attached to the stay, and the leach and foot as before.

In designing head-sails this question of one sail interfering with the work of another is of great importance. In a plan where the head area is divided into two sails this question should receive great consideration. In our older plans of boats, designed and built fifteen or twenty years ago, when the double-head rig or cutter rig first came into use in this country, we at first had a great deal of trouble in getting satisfactory head-sails, and the greater part of the trouble was from the cause I have just mentioned.

On a single-masted boat the head-sails are called forestaysail and jib, but properly speaking they are really foresail and jib, as the name forestaysail should really apply to the first sail forward of the masts on a schooner. In either rig, sloop or schooner, the two sails must be so arranged as to have the least interference possible with each other. Care must be taken to keep the jib from coming too close to the forestaysail, and one way to succeed in this is to be careful not to get the points at which the two sails attach to the mast too close to each other.

The jibstay should be a very considerable support to the mast-head. And the forestay, while being kept as much as possible clear of the jibstay, should not be placed so low down on the mast-head that the strain of the halyard in setting the sail may be in any way likely to assist in pulling forward that part of the mast near the jaws of the gaff.

The boom should be placed high enough from the deck so that the helmsman may have a good opportunity to see out under it at almost any angle of heel. On a 21-foot raceabout 18 inches is a convenient height, and at the taffrail 3 feet should make a good convenient clearance, and also keep the end out of the water when running with a free wind. Now, in getting the right length of

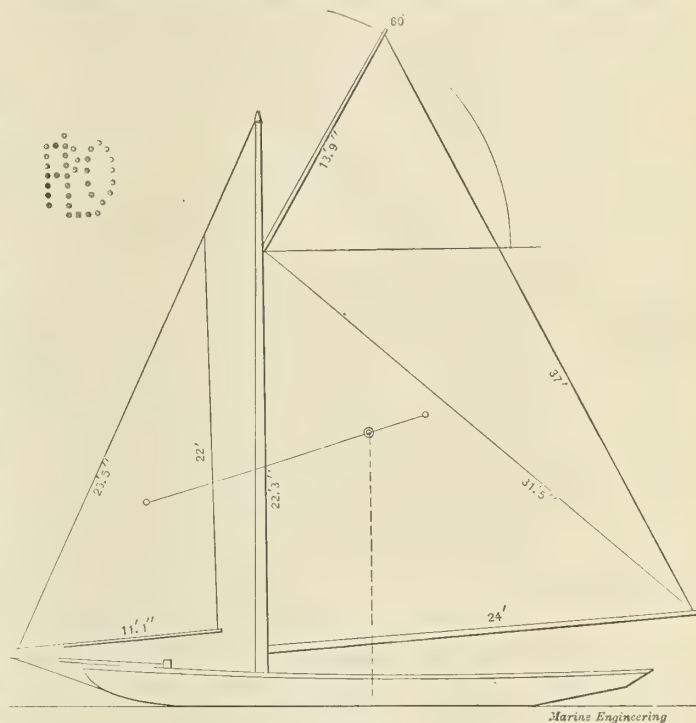


FIG. 2.

mast for a boat of this class, or any class (as this rule will invariably apply in all cases), do not lose sight of one very important point: the lower peak halyard block should never be lower than a level line drawn through the center of the gaff. This will always insure your being able to hold the peak up to its proper position. The lower peak angle will, of course, have a greater tendency to thrust or push against the mast at the jaw of the gaff than the higher angle. Should this occur it can be remedied by a strut and stay on the forward side of the mast, which has been adopted on almost all classes of racing yachts to-day, from the 15-footer to the 90-footer, and with this arrangement of rigging the spars can be made much lighter in weight.

The jib on a 21-footer is of the following dimensions: Luff, 31 feet 5 inches; leach, 22 feet; and foot, 11 feet 3 inches; the intersection of the foot and leach being the point nearest to the mast, this distance on the plan I am discussing at this point being 2 feet 4 inches and the head 3 feet from the mast; so you may readily see the care taken to prevent the jib throwing any wind under the lee side of the mainsail. Of course we would not carry this out on a larger boat, as it would not be practical on a sloop or schooner, especially one of size. Possibly I might have covered this point in fewer words by saying that in planning your head-sails do not make them lap one over the other any more than possible. I firmly believe that the best jib is one that does not lap by the forestay at all. In head-sails the foot should be cut high, the forestaysail well up and clear from the deck. Also, the jib should have a high clew (this is the point where the leach and foot intersect). The nearer a jib on leach and foot approaches the same dimensions, the more perfect will be the sail, and especially is this true of jibtopsails. While all of our knowledge on this matter has been gained in a practical way, it has been almost

from a point as near as possible to the jaw of the gaff, leading aft to hold against the forward thrust of the jaw of the gaff. The jibstay and upper shroud should hold the mast from coming aft. The lower shroud and runner should hold the mast from coming forward.

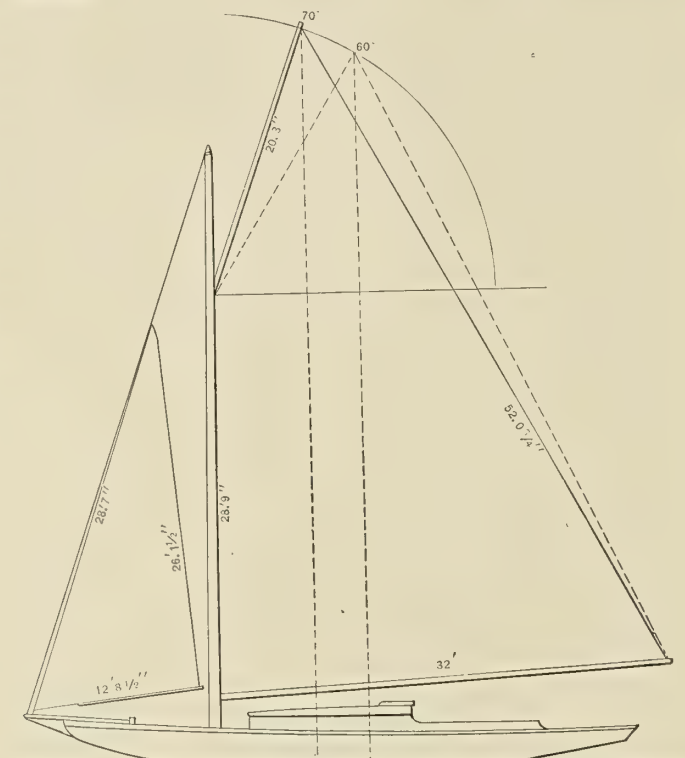


FIG. 4.

Another important point in relation to your sails is to properly trim them. Knowing after a trial at about what angle the main boom should be, great care should be taken to get the head sheets so arranged that they will work to the best advantage with the mainsail. The head sheets should not be too broad or too flat,

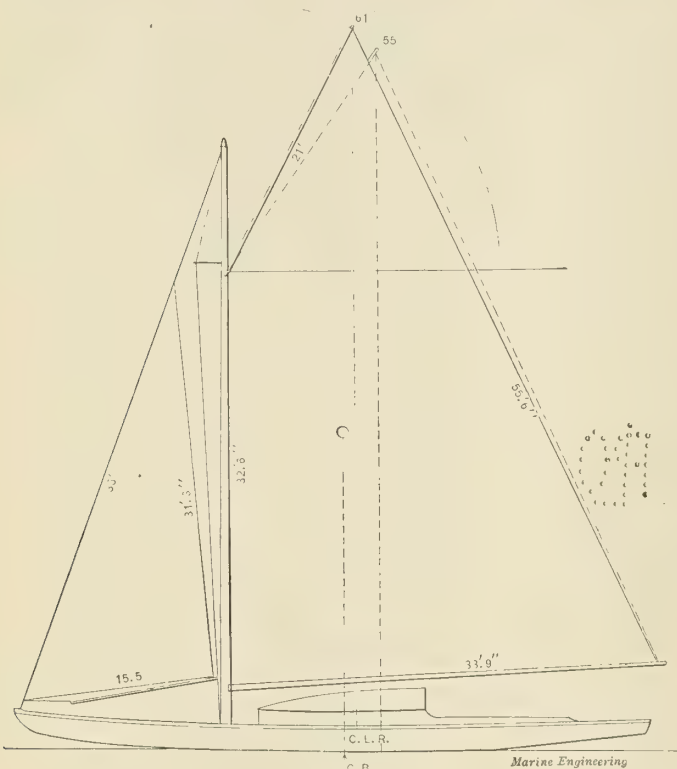


FIG. 3.

the work of a lifetime to educate yachtsmen and designers to these facts. In making plans for schooner rigs it has become almost a custom to place the mainmast at about the midship-section, which on most of the modern plans will be found near the center of the water-line. The foremast will come nearly half way from the mainmast to the stem of the vessel, and on almost all of our schooner plans I find the center of effort will be found just about at the forward side of the mainmast, and the center of lateral resistance from 18 inches to 2 feet aft of this.

The custom of making the mainmast longer than the foremast I also find falling into disuse, and on almost all of our new plans you will see the two masts of about the same height, with some little difference in lengths of the topmasts. The rigging plan should also be a part of the sail plan, as it is most important that the mast and other spars be held as straight and firmly as it is possible to make them. As soon as a mast or boom or gaff begins to bend or buckle out of shape, just so soon is your sail out of shape and losing its efficiency. The shrouds should be placed one from the top of the mast-head leading forward of a center line of the mast to hold the mast-head from springing back, and one

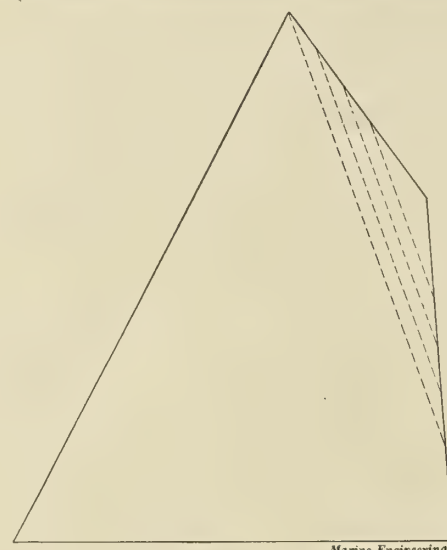


FIG. 5.

and in planning them care should be taken that their shape and proportions are so arranged as to have them sheet at convenient places on deck, or, in the case of the jib, on the rail. Many a boat has never shown her best speed for the reason that her head-sails were improperly shaped or not trimmed at the right angle for their best work with the mainsail.

In considering the weight of canvas for the different classes, it will depend, as I stated in the beginning, a great deal upon the

conditions under which a boat will be used. There is a liability of getting the canvas too light for the best work. In the extreme skimming-dish type of boat the lighter the canvas the better, but in almost all other classes the tendency is to get the canvas too light. The yachtsman, from possibly a mistaken idea, thinks his

In the past few years some radical changes have been made in the arrangement of cloths in cutting sails. It was the custom for many years for all sails to be made with the seams or cloths parallel with the after-leach, but in recent years we have been making sails with the seams or cloths at right angles to the after-



U. S. S. HARTFORD UNDER SAIL, SHOWING LARGE SINGLE TOPSAIL.

(Photo Copyright, 1901, by Enrique Muller.)

sails should be extremely light for light winds, and the sailor or skipper wants them light so he can handle them more easily. If the matter is left to the sailmaker he will seldom err by getting them too light, as he knows that in using a good weight of material his work will hold in shape longer.

leach. In some respects this is a great improvement, as undoubtedly it does away, to a great extent, with the stretching of the sail. The cross-cut sail, as we call the new style, stretches but very little in comparison with the old method, but it is not so strong and will not stand the hard usage of the old arrangement

of cloths, and consequently will not wear nearly so long a time. The cross-cut sail is not a new invention at all—it was in use many years ago. In fact, the famous sloop *Maria* had her sails made with this arrangement of cloths.

The reason for these sails not stretching so much can be readily seen if we consider the weave of a piece of canvas. It is composed of a plain weave, or a basket weave, as the weaver would call it, lengthwise of the goods in the warp, and the cross-thread is the filling. The filling thread shows almost no corrugation from the action of the loom; this is all in the warp. In weaving a piece of duck the weaver counts on a take-up of 30 percent in the warp and in the filling only 10 percent. This shows the amount of corrugation in the warp thread and is caused by the warp thread passing up and down over the filling thread. Also, there is a great deal more twist put into the warp than in the filling. Now, as soon as a strain is put onto this fabric, if put on lengthwise of the cloth the corrugated thread gives and you get a certain amount of stretch. But if this same strain is put onto

The many styles or ways of arranging the cloths in sails are too numerous to mention. Among the old sail plans filed away in the loft of my father, R. H. Wilson, of Port Jefferson, N. Y., we find plans dating back to 1836, and, to our surprise, among these we found plans dating from 1836 to 1846 that in topsail and headsails showed where the cloths had been cut, some at right angles to the leaches and others with cloths at right angles to the luffs, but there were few of the latter. One of our men who worked in the loft at that time remembered that it was then the custom, but was abandoned on account of not having sufficient strength in headsails. Numberless arrangements of cloths have been tried, some with success; others after trial were found to have had no value. While it may seem to be a fact from observation that our sails of the present day have cloths at right angles with the leaches, this is not the fact, as the cloths or seams are not cut at right angles to the after-leach. The same arrangement of cloths has been tried on square sails on ships, but not with marked success on account of the sails not wearing so long. It was possible



A BARKENTINE UNDER FULL SAIL.
(Photo by N. L. Stebbins.)

the filling thread there is but very little if any give on account of the small take-up, and so, practically, no stretch. In cutting our sails cross-cut we arrange our goods so that the least stretch comes in line with the greatest strain, and as so much of our work is handled by amateurs this arrangement of cloths is much easier for them than the old arrangement, where there was so much stretch to be taken care of. Thus the cross-cut arrangement simplifies the matter for all concerned.

The cross-cut arrangement is a great improvement in the headsails. In the forestaysail, jib, or jibtopsail made with the old arrangement of cloths, on the side next to the luff the cloths ended with a long diagonal or gore. This side of the sail was constantly stretching and was the cause of much trouble by the sail being constantly liable to stretch out of shape, and was caused by the fact that the warp thread—the one having so much corrugation—came nearest to the line of strain and consequently stretched more than if the goods had been so arranged as to bring the filling thread in the line of strain. Also, in the cross-cut sail, the diagonal ending of the cloth will of course be a much shorter line and less liable to stretch. In applying this arrangement of cloths to all headsails and topsails we have found it to be of great advantage.

that in headsails no gain was made in going over to the arrangement of cloths parallel to the leaches, as in some of our large coasting schooners the cross-cut arrangement in headsails is found to be of advantage. In these large four-, five-, and six-masters, it is a rather common occurrence to part a jib-sheet, and many times before the sail can be taken in it is so badly torn as to make it almost an impossibility to repair it; but with the cross-cut arrangement it is not liable to tear so badly but that it can be repaired at a small expense.

All sails made cross-cut become too hard and flat after use, and what stretch does take place tends to put the sail out of business for racing; that is, the longer it is used the poorer it becomes. On the contrary, the sail with parallel cloths, if properly made at first, becomes better the longer it is used; and it will hold good in shape and draught until the cloth becomes entirely useless and has lost all of its life.

In addition to what I have said above, I wish to give a few words of advice in regard to the handling of the sails after they leave the sail loft; for it is a fact that most of the poorly-setting sails that we see on yachts are the direct fault of those who have the handling of them. More poorly-setting yacht sails are made on yachts than in sail lofts. After the sailmaker has finished his

work, which he has undoubtedly done in a conscientious manner and to the best of his knowledge, if the sail is not handled in an intelligent manner the work of the maker is of no value; and unless it is handled as it should be, certainly no blame should be attached to the sailmaker.

A yacht sail, in a great many respects, is a wonderful creation, and if you will stop and consider under how many different conditions it is used it seems wonderful that so many of them retain their shape and effectiveness as long as they do. Under the varying conditions of wet and dry, hot and cold, hauled out on their spars to the limit when dry and then allowed to become thoroughly saturated with fog or rain and subjected under these conditions to use in possibly a gale of wind, and when you consider that this area of a thinly-woven material can stand usage of this kind, you must realize that in order to do it great care and study must be used in its construction to enable it to do all these things and

attach a tag to each sail, giving its dimensions, and directions as to bending and setting. If these directions are followed and the sail is faulty, either the sailmaker is at fault or some outside reason is to be found. Possibly the spars may not be of sufficient size to stand the strain, and if so should be corrected at once. No sail can set properly on crooked or buckling spars. The man who made the sail knows at what dimensions it should bend. The directions should caution the yachtsman against pulling the sail to the full dimensions in case of damp or cold weather. The duck will contract in cold and especially damp weather; but, if the conditions are favorable, pull the sail to dimensions given on the tag. After the sail is bent onto the spars it should be carefully set as follows: See that on the luff or hoist the sail is hoisted to the tag measurement, and when setting up the peak set it up so that the sail wrinkles well under the jaws of the gaff, as per sketch. (Fig. 5.)



SCHOONER YACHT CORONA, SHOWING CROSS-CUT SAILS.
(Photo by W. B. Jackson, Marblehead.)

still remain a good, useful sail. For years our time and most earnest efforts have been devoted to these very important points, and in working out these points and creating a sail that shall be both useful and also an object of beauty, and a part of the beautiful creation which every yacht should be, we have learned many things and gained a knowledge that should put us into a position to give advice to our customers by which they may be enabled to handle their sails with that intelligence that will tend to obviate many of the difficulties with which the amateur, and, in many cases also, the professional, has to contend. A yacht sail should be two things in one. It should, first of all, contain those elements that will give to the yacht the greatest amount of speed; and it should be a thing of beauty. To the customer can be given a sail that will stand simply as a picture sail, one that while it may be most pleasing to the eye has not the "go" in it that it should have. This is an easy problem to solve; but when we make a sail that will be pleasing to the eye and at the same time develop the greatest amount of speed, then we have to draw on a store of experience and data gained by long practice and a careful working out of details. In sending out new sails it is our custom to

This may seem to the man setting the sail to be an exaggeration or an extreme. Not so, for as soon as the yacht is gotten under way the peak halyard begins to give and the peak will slack down. The idea of setting the sail until it is perfectly smooth so that it shows no wrinkles is wrong, for the peak will slack down, and this must be taken into consideration. At first setting, if the day is damp or rainy, the sail should be eased in on the foot and head enough to relieve the tension caused by its shrinking from the dampness.

In setting a topsail great care should be taken to know that the clew of the sail is on a line drawn at a right angle from the topmast to the end of the gaff. This can best be judged by getting off some distance from the yacht in a small boat, as it is impossible to judge from the deck. Careful attention given to this detail will save a great deal of annoyance to the yacht owner, as topsails are the sails most criticised, and all on account of not giving this one fact proper attention and consideration. The head-sail depends entirely on the amount of strain put onto the halyard in setting, and this needs careful and good judgment. If your sail is set too hard on the luff it will invariably have a tight leach.

If not set hard enough it will have a slack leach. A warm, pleasant day, with light winds, is most to be desired in setting new sails the first time. Do not use them at first in wholesail or reefing winds, as it is very much against the future development of the sails. The yacht had better lie at her mooring for a week waiting for the best conditions than to spoil the sails the first time they are used. The best sails may not look well at first setting—they may seem baggy and full of wrinkles—but after being underway a short time under favorable conditions they will work themselves into a better shape. Ninety-five percent of the sails we make are used by amateurs, and we have to make the sails to counteract the handling they get, and I think it is astonishing to see the amount of good results we accomplish in working to overcome these conditions under which our work is used. So it is a great gratification to us to know that no better sails are made than the American racing sails. In fact, I do not hesitate to say that the American racing sails are the best produced in the world. Our material is the best, although it has been a hard fight on our part as sail-



KNOCKABOUT, WITH CROSS-CUT SAILS.

(Photo by W. B. Jackson.)

makers to get it up to its standard. Many different grades of fiber have been tried and experimented with to produce the best duck for use here in our climate. We have used Sea Island, Peeler long staple, Texas, Mississippi, Egyptian, Ramie, and, lastly, pure white silk. Of the grades of cotton used we have found that the long staple Texas will make the best grade of carded goods, but for our fine racing canvas a long Peeler of Texas or Mississippi grade makes a better and firmer grade of cloth than the Sea Island or Egyptian. The silk we have used only in very light balloon sails for cup defense. We have worked these questions out until for our own use we have our cloth manufactured in one mill which weaves for us a cloth which, in its construction, is the result of our own experience.

So, the reader may see that in order to produce a first-class yacht sail for racing purposes, one that we say will be a thing of beauty and at the same time give the best results as to speed, is no haphazard affair, but the result of long and careful study and experience, to which we have added the most careful working out of every detail.

The Amstelstroom in Collision.

The steamship *Amstelstroom* had a collision with the steamship *Cuxhaven*, of Goole, England, just outside the docks of Goole on the 6th of last October. The *Amstelstroom* was waiting opportunity to enter the docks, while the *Cuxhaven* was just on the point of leaving port. When passing the *Amstelstroom* the *Cuxhaven's* steering engines refused to operate, causing her to take a heavy sheer to starboard and to collide with the *Amstelstroom*. The photograph shown exhibits the damage done.



THE DAMAGE TO THE STEAMSHIP AMSTELSTROOM.

(Photo by James Fisher, Hull.)

The *Amstelstroom* is owned by the Hollandsche Stoomboot Maats, of Amsterdam, having been built at Tynewood in 1885. Her dimensions are 222 feet 7 inches in length, 28 feet 4 inches beam, and 14 feet 7 inches depth. She has four-cylinder, triple-expansion engines, the dimensions being 30-inch, 42-inch, 60-inch, and 60-inch cylinders, with a common stroke of 54 inches. There are two double-ended boilers having a combined heating surface of 7,444 square feet.

The Department of Commerce and Labor reports that during the six months ending December 31, 1904, there were constructed in the United States 542 vessels of a total of 92,598 gross tons, of which total 509 vessels of 72,464 tons were wood and 33 vessels of 20,134 tons were steel. Of the total, 227 vessels of 60,885 tons were propelled by sail and 315 vessels of 31,713 tons were steam.

Comparing the total with that for the six months of the preceding year, we find that it has fallen off almost one-half; the number of ships in the preceding period having been 571, and the gross tonnage 177,067. Totals are as follows:

Calendar	year ending	Dec. 31,	1901,	1,322	ships,	376,129	gross tons.
"	"	"	"	"	"	1902,	1,252 " 429,327 " "
"	"	"	"	"	"	1903,	1,159 " 381,970 " "
Fiscal	"	"	June 30,	1904,	1,094	"	349,573 " "
Calendar	"	"	Dec. 31,	1904,	1,065	"	265,104 " "

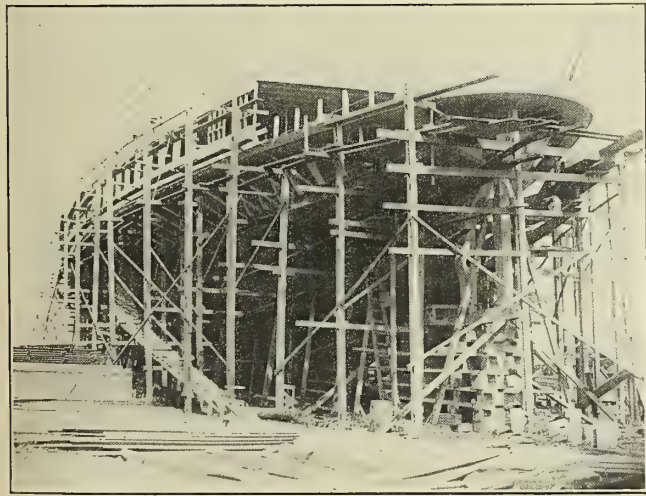
NEW FERRYBOATS FOR SAN FRANCISCO HARBOR.

The demand for a fifteen-minute ferry service between Oakland and San Francisco, a distance of some three miles, has led the San Francisco, Oakland, and San Jose Railway Company to contract for two ferryboats of a somewhat novel pattern, capable of a maximum speed of 17 miles per hour, and of a high power of acceleration. Experiment has shown that the smallest interval of time which can be allowed for in the slip is about 3 1-2 minutes, and, as there are only two of the boats, this makes it necessary that the trip should be made in from 10 to 11 1-2 minutes.



THE SAN FRANCISCO ON THE STOCKS.
(Photo, W. S. Atkinson.)

For the accomplishment of this result, Mr. Henry J. Gielow has produced the following design: A boat with a length over all of 201 feet, a length of keel of 176 feet, a beam, molded, of 35 feet 3 inches, a total width over the guards of 58 feet, and a depth of hold amidships of 17 feet 9 inches. The boats are built of wood, the material being Puget Sound yellow fir, except that the keel, stem, and stern posts, rudder post, and frames are made of white oak, the keel being 12 inches square and the frames 6 by 14 inches,

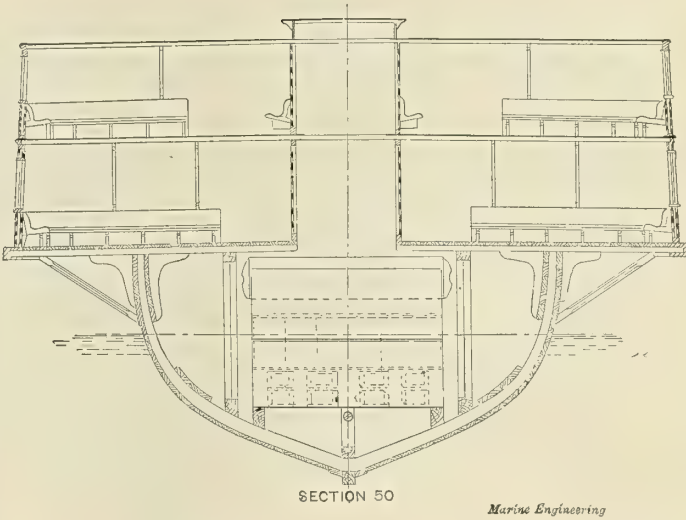
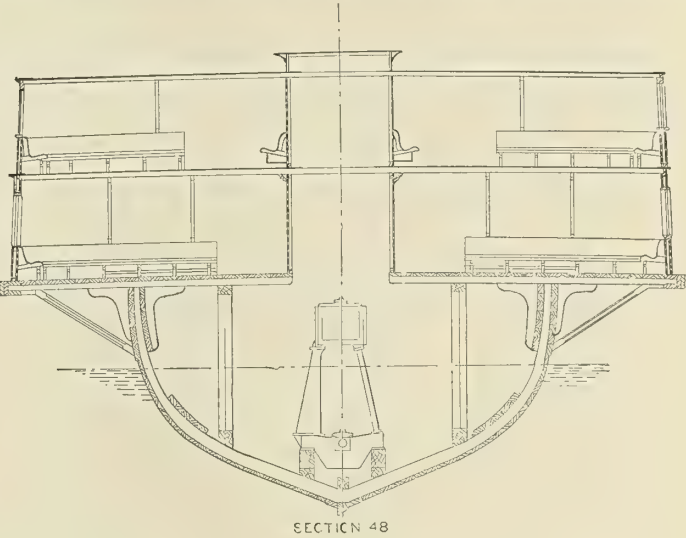


THE SAN FRANCISCO ON THE STOCKS.
(Photo, W. S. Atkinson.)

spaced 24 inches center to center. The main keelson is of pitch pine, 12 inches square, and, as with the keel, is to be fitted in not more than four pieces, from end to end of the ship. The deck beams measure 5 by 7 inches and are fitted to every frame. The outside planking is as follows: sheer strake, 4 by 12 inches; the next six strakes, 4 by 10 inches; garboard strakes, 4 by 12 inches; and the intermediate strakes, 3 inches in thickness and from 8 to

10 inches in width. There are five bulkheads located at frames 12, 25, 49, 68, and 86, thus dividing the ship into six compartments. The bulkhead on frame 49 is recessed for a thrust bearing and fitted with a hinged water-tight door. The hull from the keel up to 18 inches above the designed water-line will be covered with 22- and 20-ounce hard-rolled copper, laid over sheathing felt. Unless the copper sheathing should be pierced this sheathing felt will make the ship at all times absolutely water-tight. The deck planking is of pine, measuring 3 by 4 inches, and fitted in the longest possible lengths, each strake being fastened with one 5-inch galvanized-iron deck spike to one frame, and two spikes to the next.

The main deckhouse is 128 feet 6 inches long with an additional projecting hood of 8 feet 3 inches at each end, making a total length of 145 feet. There will be clear head-room under the carlines of 8 feet 6 inches. On top of the main deckhouse is a saloon deckhouse 86 feet in length, and having a clear head-room

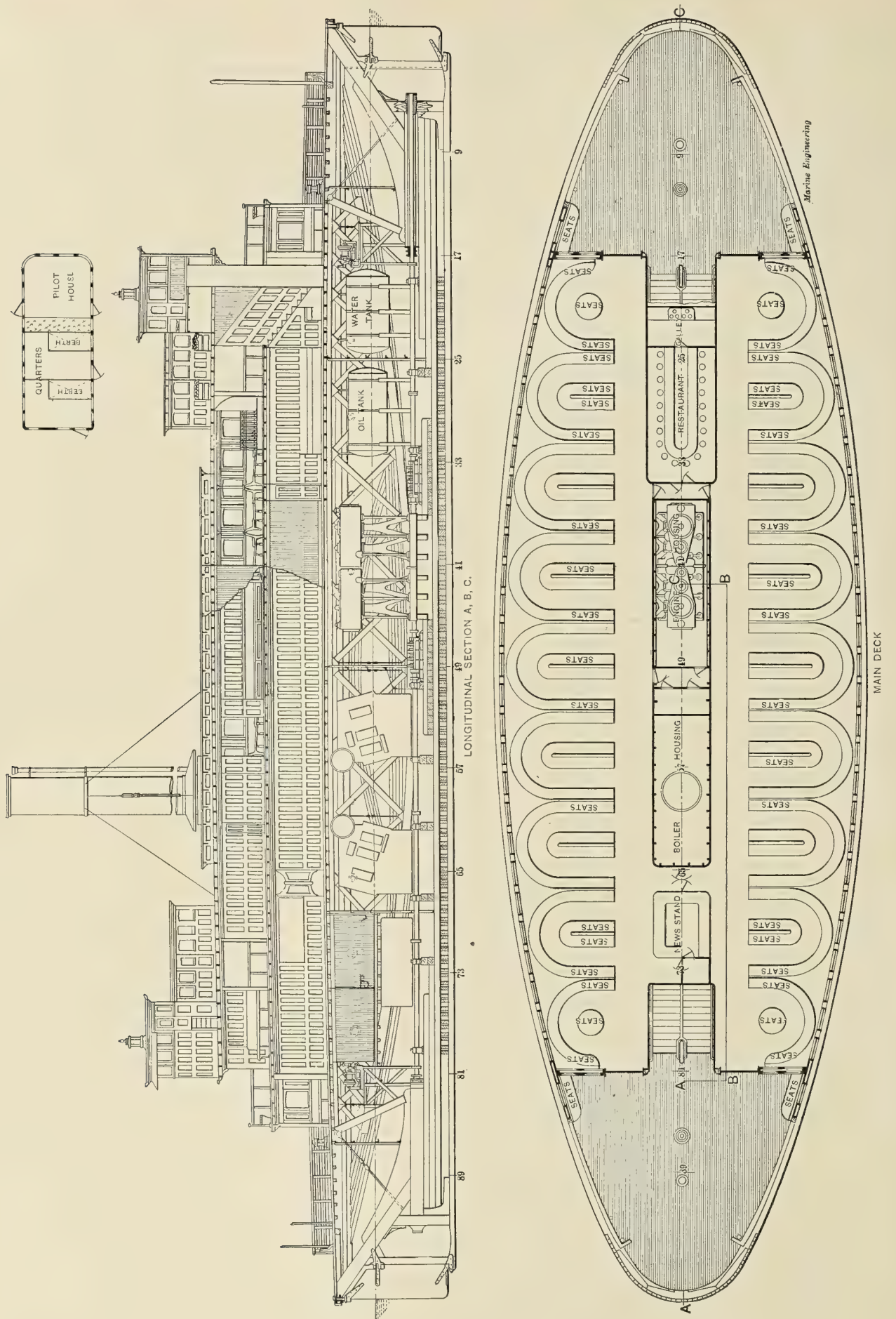


Marine Engineering

of 7 feet 3 inches at the side and 7 feet 6 inches in the middle of the vessel. There are two pilot houses, located one at each end on top of the saloon deckhouse. These are each 12 feet long by 10 feet 6 inches wide, and give a clear head-room of 7 feet.

On the main-deck, in one end of the cabin, will be a news stand finished in mahogany, with necessary racks and shelves. Extending along the center line of the vessel will be light and ventilating shafts for boiler room and engine room, a restaurant and galley, and a barber shop, it being estimated that a man will have just about time for a shave during the passage.

One peculiar feature about these boats is that teams will not

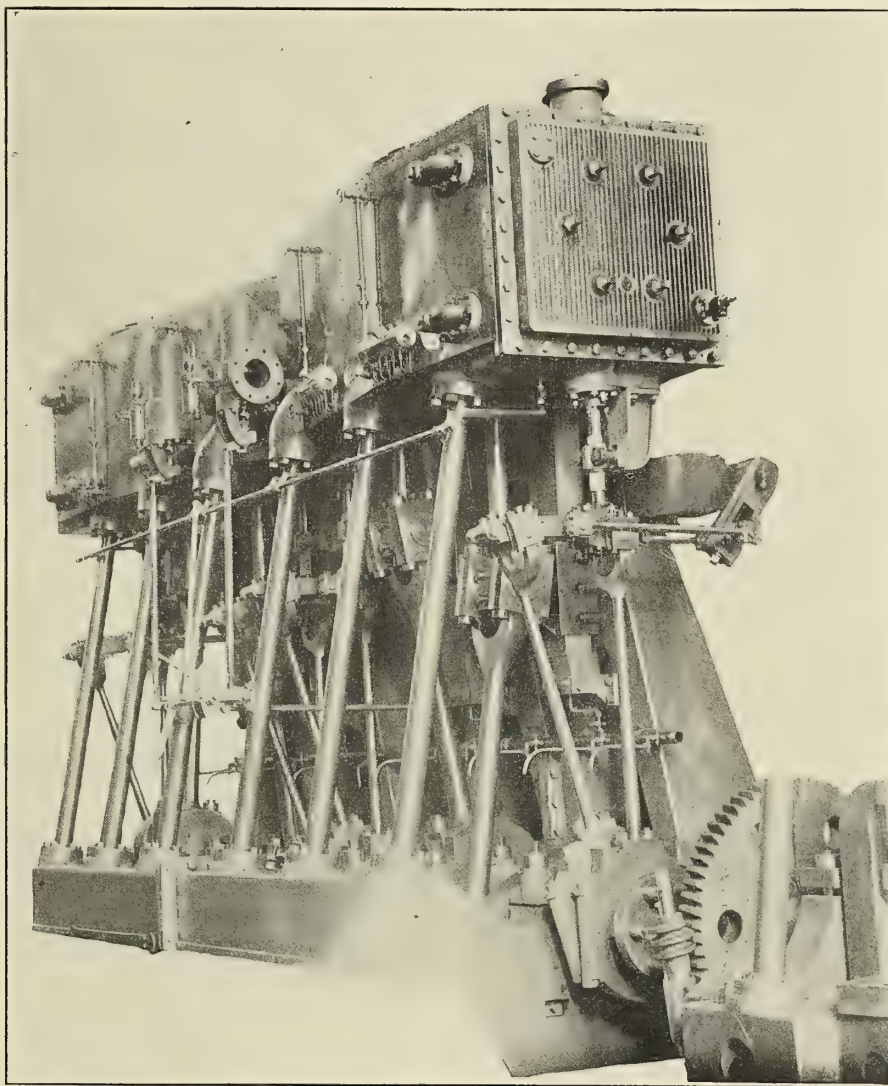


be carried, provision being made simply for passengers. This has resulted in an odd arrangement of the seats, as shown by the main-deck plan. The saloon-deck has just the same arrangement of seating as has the main-deck.

There is one engine in the form of a double-compound engine, having two high-pressure cylinders 20 inches in diameter, and two low-pressure cylinders with a diameter of 42 inches; all having a common stroke of 28 inches. This engine actuates a shaft carrying a screw at each end of the ship. The engine will be operated under a working pressure of 160 pounds per square inch at the throttle, at which pressure it will run at about 150 revolutions per minute. This gives a piston speed of 700 feet per minute. The indicated horsepower at full speed will be about 2,000. The pistons are cast steel, conical in shape, and fitted with cast-iron packing and bull-rings. The followers are also cast steel. The

ing a pitch-ratio of 1.29. The approximate blade area is 35 square feet, this providing ample surface for a rapid acceleration of the boat.

The two boilers are of the Babcock and Wilcox type, designed for a working pressure of 200 pounds per square inch, each being 10 feet 1 inch long, 16 feet 1 inch wide, and 12 feet 8 inches in height. The length of the steam drum, including its heads, is about 17 feet 6 inches. The tubes are of 2-inch and 4-inch diameter, the former having a thickness represented by No. 10 B. W. G., and the others, No. 6 B. W. G. The steam and water drums are 42 inches in diameter, and are made of open-hearth steel plates 1-2 inch in thickness. The drum heads have a thickness of 9-16 of an inch. The stack extends 48 feet above the main-deck, and is doubled, having an inside shell 68 inches in diameter, and constructed of No. 10 steel, and an outside shell of No. 12



THE ENGINES OF THE FERRYBOAT SAN FRANCISCO.

piston rods are machinery steel forgings 4 inches in diameter. Stuffing boxes for piston rods and valve stems are fitted with Katzenstein metallic packing. High-pressure valves are of the piston type, 10 inches in diameter, and single-ported. The low-pressure valves are balanced, double-ported slide valves made of cast iron. The valve gear is of the Stephenson type, with double bar links, all valves being worked direct. Connecting rods are of machinery steel, and are 70 inches long between centers. The shaft is of forged machinery steel, 8 1-2 inches in diameter for the intermediate sections and 9 inches for the tail sections. The propellers are two in number, of manganese bronze, four-bladed, with a diameter of 9 feet 9 inches, and a pitch of 12 feet 6 inches; giv-

steel 80 inches in diameter. It is stayed by six galvanized steel wire-rope guys 2 inches in circumference. There are a pair of Blake vertical simplex pumps, having steam cylinders 12 inches in diameter, water cylinders 7 inches in diameter, and a common stroke of 12 inches. The water cylinders are to be tested to a pressure of 300 pounds per square inch. For fire protection, a Blake horizontal duplex pump is fitted, having steam and water cylinders 10 inches and 7 inches respectively in diameter and 12-inch stroke. The circulating pump is centrifugal with 12-inch suction and discharge pipe, and is driven by a compound engine with cylinders 5 and 10 inches in diameter and 8 inches stroke. The air pump is a Blake, single-acting, twin vertical pump, with

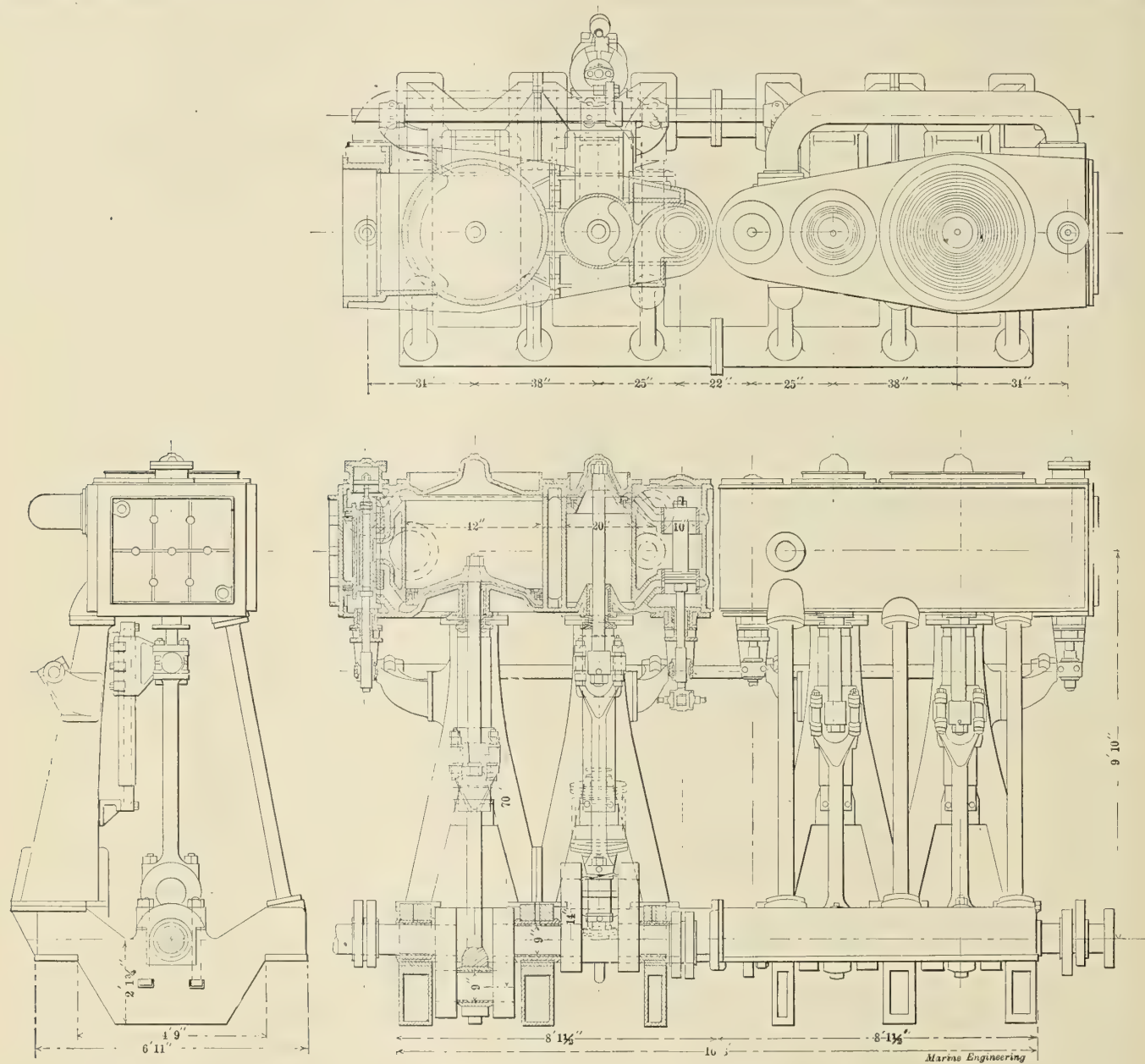
two steam cylinders 9 inches in diameter and two air cylinders 18 inches in diameter, with a stroke of 12 inches.

The boilers are fitted for burning fuel oil, which is carried in two tanks, as shown in the inboard profile. This oil is pumped to the furnaces by a double pumping system consisting of two Blake horizontal duplex pumps, having steam cylinders and oil cylinders, respectively, 4 1-2 inches and 2 3-4 inches in diameter, with a common stroke of 4 inches. Steam will be used for atomizing the oil. The oil tanks are each 6 feet in diameter and 12 feet in length constructed of steel 5-16 inch in thickness. The combined capacity is about 5,000 gallons.

Ocean Yacht Race.

The Kiel Yacht Club has set May 15 as the date for the beginning of the ocean yacht race for the cup which the emperor of Germany has offered. No restriction as to nationality is provided, but all boats competing must have a tonnage of at least 200. There are only four or five German yachts which fulfill this tonnage requirement, of which two belong to the emperor himself, one being the *Meteor III*, which was built in the United States a few years ago.

The management of the race on this side of the ocean will be



THE ENGINES OF THE FERRYBOAT SAN FRANCISCO.

The boats are lighted by electricity generated by two direct-coupled sets each having an output of 15 kilowatts at 400 revolutions and 110 volts. An automatic governor is to be fitted so that a variation of 25 percent in the load will not affect the speed of the engine more than 2 percent. There will be about 360 lamps, nearly all of 16 candlepower, and required to develop 1 candlepower for each 3.1 watts furnished.

The hull and joiner work have been constructed by John W. Dickie, of San Francisco; the boilers at the Bayonne shops of the Babcock & Wilcox Company; and the engines and shafting by John W. Sullivan, of New York. The boats will be ready for a trial trip early in March.

conducted by the New York Yacht Club. It is expected that several American yachts will enter the competition, and also that some English, and possibly one or two French, yachts will compete. There has been some talk of changing the rig of the cup defender *Columbia*, so that she will be able to enter a race of this character. There is some idea that if this were done she would stand a very good chance of coming in near the head.

The emperor's idea is to get American yachts to the other side in order to institute comparisons between them and German yachts, and thus give something of a boom to yacht racing in German waters. It is reported that several American yachts are already entered.

PRACTICAL POINTS ABOUT THE SCREW PROPELLER.

BY W. F. DURAND.

PART III. DESIGN.

The problem of propeller design is concerned primarily with the determination of such dimensions, form, and proportions of propeller as shall best fulfill certain specified purposes, the latter being stated in terms of some combination of power, speed, and characteristics of the ship. The usual data which are furnished as points of departure are speed and power, and the design is commonly based on these two items, with such additional aid as may be derived from a knowledge of the ship. It results that in all formulæ and methods commonly employed for this purpose, horsepower and speed are primary items, combined with the controlling features of the propeller and of its operation. Where the constants employed in such formulæ are based on experimental determination, and where the formula or method is intended to take account of all the important controlling items which may enter into the problem, such method is undoubtedly the most suitable, providing as it does for such detailed analysis of the probable performance as the data in hand will permit of making, and for due recognition of the various items which may enter as factors into the final result. It is safe to say, however, that actually but few propellers are designed with reference to such detailed analysis, and that most of the formulæ which have been used for these purposes are admittedly quite approximate in character, and must of necessity contain a large element of judgment. There is always room for a formula or method of this approximate character, especially if it is reasonably simple in use and if it provides at the same time for some approximate recognition of the effects due to the principal variable items on which the performance as a whole will depend.

With these purposes in view the method of design given in the present chapter has been developed. It should be understood at the start that this method is not offered as a substitute for those of a more detailed character and including more complete recognition of the various items, but rather as a convenient and simple method for the guidance of those who are not experts in this field of design, but who may wish to determine the dimensions of a screw propeller with some assurance of a reasonably satisfactory result.

In proceeding with the development of the method we may first call attention to the fact that within the usual limits of admissible error we may assume the I. H. P. required for the propulsion of the ship to vary as the cube of the speed, especially with the provision of a corrective factor for controlling departures from this general law. Furthermore, it will be found that in all propeller formulæ, however disguised, the power absorbed by the propeller is assumed to vary likewise as the cube of the speed. Accepting these relationships it should be possible to eliminate the item of power, and to thus develop a relation directly between the functions of the ship and those of the propeller on which the item of power depends. This would result in relating the propeller directly to the ship and independent of the power, except indirectly, as will later appear. That such a direct relation between the propeller and the ship is reasonable may also be seen by remembering that the same propeller must in fact be used often over a wide range of power and speed, and that if well designed it will operate over such range with but slight variation in efficiency. It is again reasonable when we remember that in effect it is the resistance of the propeller which overcomes the resistance of the ship. The propeller is rotated at the stern of the ship, the blades meet with resistance to their motion, and the fore-and-aft component of this resistance furnishes the propulsive thrust. There is therefore a balance between the resistance of the ship and the fore-and-aft component of the resistance of the propeller, and such being the case it seems reasonable to assume that the size of the propeller may be directly related to the size of the ship, the details of the relationship being made to include such special items of the case as the data in hand will permit. A further reason

for the convenience of a formula of this character is found in the fact that oftentimes, especially with the non-expert designer, the size of the ship or boat is known independent of the power or speed. The problem is simply one of fitting to a given ship or boat a propeller such that whatever the power it will be economically absorbed for propulsive purposes, and such speed realized as may properly be expected under the conditions.

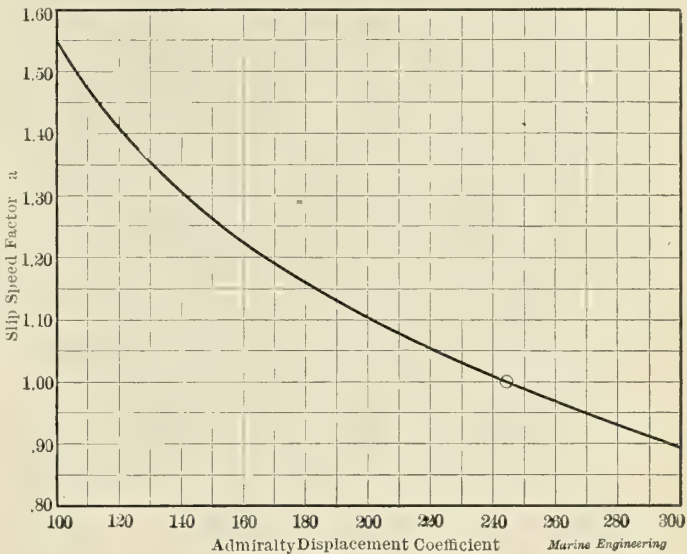


FIG. 34.

With this, by way of general explanation, we will state the formula at once, and proceed with its discussion by means of illustrative problems.

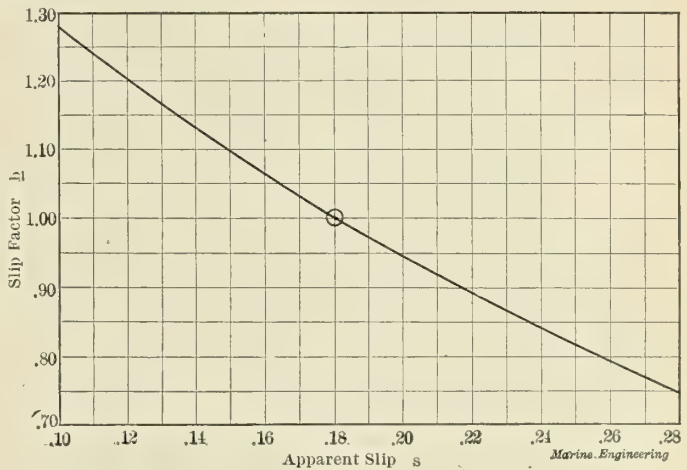


FIG. 35.

- Let D = displacement of ship in tons, or in case of twin or triple screws, total displacement divided by number of shafts.
- u = speed of ship in knots.
- v = speed of ship in miles per hour.
- d = diameter of propeller in feet.
- p = pitch of propeller in feet.
- N = revolutions per minute.
- a = ship-speed factor derived from Fig. 34.
- b = slip factor derived from Fig. 35.
- c = pitch-ratio factor derived from Fig. 36.
- e = area factor derived from Fig. 37.
- s = apparent slip-ratio.

Then we have:

$$d = abce \sqrt[3]{D} \dots\dots\dots (1)$$

$$N = \frac{101.3u}{p(1-s)} \text{ or } \frac{88v}{p(1-s)} \dots\dots\dots (2)$$

$$s = \frac{pN - 101.3u}{pN} \text{ or } \frac{pN - 88v}{pN} \dots\dots\dots(3)$$

Again, let d = diameter of propeller in inches.
 W = displacement of boat in pounds, or total weight in pounds, of boat and load in normal condition.

Then we have:

$$d = .9abce \sqrt[3]{W} \dots\dots\dots(4)$$

where a , b , c , and e have the same meaning and values as in (1).
The fundamental equation (1) means simply that the diameter of the propeller is equal to the cube root of the displacement in tons, multiplied by a combination factor a , b , c , e , which, as explained later, is to be derived from the special condition of the case. The diameter being thus determined, the pitch, revolutions, and surface area will directly result from the various assumptions required for the determination of the factors b , c , and e . On the other hand, if the diameter, pitch, or revolutions are fixed, some one of the factors b , c , or e is left to be determined by the fundamental equation, and the possibility of a satisfactory solution with the assumed values is determined by the resulting value of such factor. These various points will be made clear by means of illustrative problems to follow.

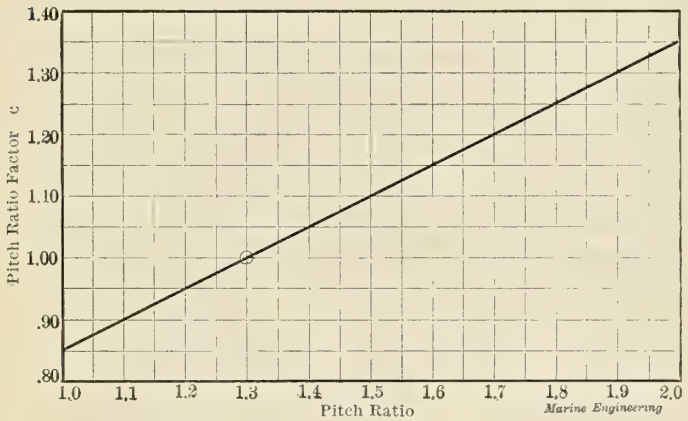


FIG. 36.

We must first, however, discuss briefly the nature of the factors a , b , c , e .

The ship-speed factor a is intended to take account of the dimensions and form of the ship, especially in relation to the expected speed. To those familiar with the subject of ship powering, it will be sufficient to state that this factor represents the Admiralty coefficient K , in the well-known formula $H = D u^3 \div K$. The value of K which corresponds to the value of $A = 1$ is 242, while the lower limit of about .9 corresponds to 300, and the upper limit of 1.55 corresponds to 100. The values of this factor are in effect $\sqrt[3]{242 \div K}$, and with this general explanation those familiar with common or appropriate values of the Admiralty coefficient will be able to readily select a correspondingly suitable value for the factor a . To those not thus familiar, some indications may be given as a general guide.

For any given ship, speeds measured in knots decidedly greater than the square root of the length in feet, may be called high or excessive. Where the speed is not much in excess of the square root of the length in feet, or when it is less than such square root, it may be considered as moderate or low. Thus, for a boat 100 feet in length, 10 to 12 knots is moderate speed, 15 to 20 or above high to excessive. For 400 feet in length, 20 knots is moderate, 25 high. With this understanding, ships of fair size, not too full in form, and at speeds not much exceeding the square root of the length in feet, will show a value of a close about unity. For larger ships or more moderate speeds, and in general under conditions favoring high propulsive efficiency, the values will fall through the lower range to about .90, or in rare cases slightly below. On the other hand, with smaller craft or full forms, and at high or

excessive speeds, the values will rise through the higher range. For very fine forms, and such excessive speeds as are found with torpedo boats and very fast yachts, the values are usually lower than might be anticipated from the relation of the speed to the square root of the length in feet. Experience alone, or still better, experience combined with a proper use of the so-called "law of comparison," can best serve to safely select values for this coefficient, but representative values may be suggested in classes as follows:

CLASS.	VALUES OF a .
Launches and small craft at relatively high speeds....	1.2 to 1.5
Small yachts at moderate to high speeds.....	1.15 to 1.4
Large yachts at moderate to high speeds.....	1.0 to 1.2
Cargo steamers, low to moderate speeds.....	.90 to 1.0
Large passenger steamers, high speeds.....	.95 to 1.0
Battleships and cruisers, high speeds.....	1.0 to 1.1
Torpedo boats and fancy yachts, excessive speeds....	1.10 to 1.2

The slip factor b is intended to provide for the increasing thrust which is obtained with increasing slip, and the consequent reduction in the diameter of the propeller. The slip is one of the conditions which must be assumed in the solution of the problem, and the value first assumed must often be modified in order to finally reach a satisfactory adjustment of the result with the various special conditions of the problem.

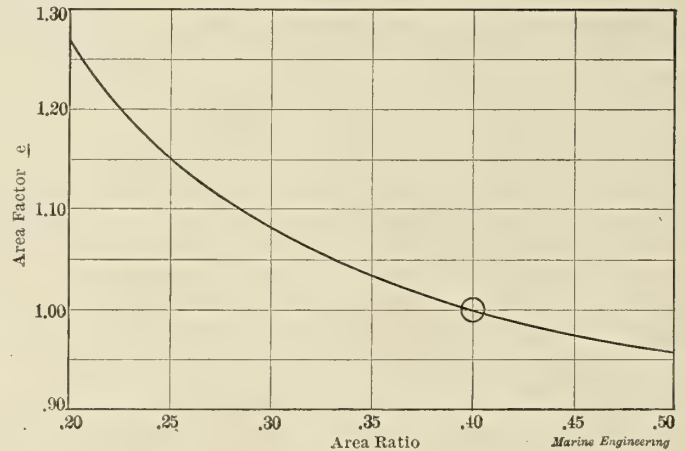


FIG. 37.

It should be noted at this point that by the term *slip* we here mean the so-called "apparent" slip as defined by equation (3). Due to the influence of the following wake in which the propeller works, the true slip is always greater than the apparent, but for simplicity we have here included reference to but one slip, and have expressed the factor b in terms of the apparent slip as most serviceable for the purposes in present view. This simplification has involved the assumption of a standard or average value of the wake influence which is supposed to apply to all cases, and of numerical value as determined by an average wake velocity of about .10 u , or 10 percent of the speed of the ship. It may be noted in this connection that while an error in the assumed value of the wake may modify appreciably the characteristics of the propeller resulting from the equations, yet the loss in actual efficiency will usually be small, and for most cases within our present purposes the average value may be safely assumed to apply, and the slip factor thus expressed in terms of the apparent slip as given in Fig. 35.

The pitch-ratio factor c is intended to take account of the fact that the higher the pitch-ratio the lower the thrust, other things being equal, and therefore the greater the necessary diameter of propeller. The selection of the pitch-ratio in any given case is often simply a matter of arbitrary choice, and the value first selected may require readjustment either to bring the revolutions within the desired range of values, or to make possible a satisfactory result on a specified diameter. Illustrations of the use of this factor will be given at a later point.

The surface-area factor e is intended to provide for the slowly increasing thrust with increase of area. The standard form of blade assumed is supposed to be elliptical or oval in contour, and the standard developed or surface area is taken as .40 of the disk area of the propeller. For such area the value of the factor e is 1, while for other values of the area-ratio the values are as given by the diagram in Fig. 37. Detailed investigation of the influence due to area shows that it depends on pitch-ratio and on slip, but for simplicity the indications of Fig. 37 are confined to average values of the slip, and are intended to apply more especially to values of the pitch-ratio, about 1.3. In all cases the same generally oval form of blade is assumed. If the blade was made broader at the tip the factor e might be taken somewhat smaller, but experimental results are lacking with reference to definite values of this influence.

With regard to the area-ratio, we may here indicate its usual values for propellers of 2, 3, and 4 blades:

Let d = diameter of propeller.

A = total developed area of blades.

Then area-ratio = $A \div \left(\frac{\pi d^2}{4}\right)$.

For 2 blades, area-ratio is usually found between .20 and .25
 " 3 " " " " " " " .30 " .40
 " 4 " " " " " " " .35 " .50

We will now proceed to the illustration of the use of these equations, by means of several illustrative problems:

(1) Given $D = 1,500$ tons.

Type, large yacht.

Propeller, one, 4 blades.

Speed, 16 knots.

Assume factor $a = 1.05$.

Take trial slip = .18 and pitch-ratio = 1.3.

Then factor $b = 1.00$

and factor $c = 1.00$.

Assume also standard area.

Then factor $e = 1.00$.

We then have immediately:

$$\begin{aligned} d &= 1.05 \sqrt[3]{1,500} = 1.05 \times 11.45 = 12 \text{ feet.} \\ p &= 1.3 \times 12 = 15.6 \text{ " } \\ N &= \frac{101.3 \times 16}{15.6 \times .82} = 126.7 \\ \text{Area} &= .40 \times 113.1 = 45.24 \text{ sq. ft.} \end{aligned}$$

Such a propeller could then be reasonably expected to develop 16 knots on 126.7 r. p. m., or any proportional speed on approximately corresponding revolutions, and in general to efficiently do its part in the attainment of any speed of which the boat and power provided will permit.

(2) Suppose in problem (1) that higher r. p. m. are considered desirable for the engine, but without fixing any definite value. Such change in the r. p. m. may be brought about either by a decrease in the pitch-ratio, or diameter, or both.

To this end let us assume as follows:

Factor a as before = 1.05.

Slip = .22.

Pitch-ratio = 1.1.

Then $b = .89$.

$c = .90$.

Take e as before = 1.00.

Then we have:

$$\begin{aligned} d &= 1.05 \times .90 \times .89 \times \sqrt[3]{1,500} = 9.63 \text{ feet.} \\ p &= 1.1 \times 9.63 = 10.6 \text{ " } \\ N &= \frac{101.3 \times 16}{10.6 \times .78} = 196 \\ \text{Area} &= .40 \times 72.8 = 29.1 \text{ sq. ft.} \end{aligned}$$

Intermediate values will, of course, give intermediate results. These two problems thus represent two widely different solutions of the same case, either of which would presumably give satis-

factory results, assuming, of course, the necessary I. H. P., and engines corresponding to the r. p. m. in each case.

(3) Given $D = 7,600$ tons.

Type, coasting, cargo, and passenger steamer.

Speed, 15.5 knots.

Propeller, one, 4 blades.

Take factor $a = .96$.

Take trial slip = .20 and pitch-ratio = 1.4.

Then $b = .94$.

$c = 1.05$.

Take $e = 1.00$.

$$\begin{aligned} \text{Then } d &= .96 \times .94 \times 1.05 \sqrt[3]{7,600} = 18.63 \text{ feet.} \\ p &= 1.4 \times 18.63 = 26.08 \text{ " } \\ N &= \frac{101.3 \times 15.5}{26.08 \times .80} = 75.2 \\ \text{Area} &= .40 \times 272.6 = 109 \text{ sq. ft.} \end{aligned}$$

The revolutions in this solution are quite moderate, and if it is desired to reduce the diameter, such reduction may be accomplished by a reduced pitch-ratio and increased revolutions. Thus, assume pitch-ratio = 1.3, and also area-ratio about .45.

Then we have in this case:

$$\begin{aligned} a &= .96 \\ b &= .94 \\ c &= 1.00 \\ e &= .98 \\ \text{and } d &= .96 \times .94 \times .98 \sqrt[3]{7,600} = 17.4 \text{ feet} \\ p &= 1.3 \times 17.3 = 22.6 \text{ " } \\ N &= \frac{101.3 \times 15.5}{22.6 \times .80} = 86.9 \\ A &= .45 \times 238 = 107 \text{ sq. ft.} \end{aligned}$$

A still further reduction might, of course, be made in the diameter by assuming a higher value of the slip with its reduced value of the factor b .

(4) Given $D = 14,000$ tons.

Type, armored cruiser.

Propellers, two, 3 blades each.

Speed, 22 knots.

Assume $a = 1.00$.

Take trial slip = .20, and pitch-ratio = 1.3.

Then $b = .94$

$c = 1.00$

Take $e = 1.00$

Then we have:

$$\begin{aligned} d &= .94 \sqrt[3]{14,000} = 18.0 \text{ feet} \\ p &= 1.3 \times 18 = 23.4 \text{ " } \\ N &= \frac{101.3 \times 22}{23.4 \times .80} = 119 \\ \text{Area} &= .40 \times 254.5 = 101.8 \text{ sq. ft.} \end{aligned}$$

A somewhat different and possibly more satisfactory propeller may be developed by taking a lower pitch-ratio and lower slip. Thus, let us take pitch-ratio = 1.2 and slip = .18, the other factors remaining as before. We then have:

$$\begin{aligned} a &= 1.00 \\ b &= 1.00 \\ c &= .95 \\ e &= 1.00 \\ \text{and } d &= .95 \sqrt[3]{14,000} = 18.17 \text{ feet} \\ p &= 1.2 \times 18.17 = 21.80 \text{ " } \\ N &= \frac{101.3 \times 22}{21.8 \times .82} = 124.7 \\ \text{Area} &= .40 \times 259.3 = 103.7 \text{ sq. ft.} \end{aligned}$$

If the diameter of this propeller must be decreased it may be done by increasing the slip and the revolutions. Thus, let us take the same values as in the last case except for the slip, which we will assume at .20. We then have:

$$\begin{aligned} a &= 1.00 \\ b &= .94 \\ c &= .95 \\ e &= 1.00. \end{aligned}$$

$$\begin{aligned} \text{Then } d &= .94 \times .95 \sqrt[3]{7,000} = 17.1 \text{ feet} \\ p &= 1.2 \times 17.1 = 20.5 \text{ " } \\ N &= \frac{101.3 \times 22}{20.5 \times .80} = 136 \\ \text{Area} &= .40 \times 230 = 92 \text{ sq. ft.} \end{aligned}$$

(5) We will next vary the problem by assuming fixed revolutions:

Given $D = 460$.

Type, torpedo-boat destroyer.

Propellers, two, 3 blades each.

Speed, 28 knots.

Revolutions, 320.

Take $a = 1.20$

slip $= .17$

$b = 1.03$.

$$\text{Then } PN = \frac{101.3 \times 28}{.83} = 3,420$$

and $p = 3,420 \div 320 = 10.7$ feet.

Take pitch-ratio $= 1.35$.

Then diameter $= 10.7 \div 1.35 = 7.93$ feet

and $c = 1.025$.

Then:

$$\begin{aligned} 7.93 &= 1.20 \times 1.03 \times 1.025 e \sqrt[3]{230} = 7.77 e \\ \text{or } e &= 7.9 \div 7.77 = 1.017. \end{aligned}$$

This indicates an area-ratio of about .37, and hence:

$$\text{Area} = .37 \times 49.4 = 18.3 \text{ sq. ft.}$$

These values indicate a practicable solution along the assumed lines, and a well-proportioned resultant propeller.

(6) Given $D = 2,000$ tons.

Type, gunboat, single screw.

Speed, 16 knots.

Suppose diameter fixed at 12 feet.

Take $a = 1.04$

slip $= .16$

pitch-ratio $= 1.4$

Then $b = 1.06$

$c = 1.05$

Then we have:

$$\begin{aligned} 12 &= 1.04 \times 1.06 \times 1.05 e \sqrt[3]{2,000} \\ \text{or } 12 &= 14.58 e \\ e &= .82. \end{aligned}$$

An examination of Fig. 37 clearly shows that this value of e is out of the question, and that in consequence the values assumed for slip and pitch-ratio must be recast. Let us therefore take new values as follows:

$a = 1.04$

slip $= .18$

pitch-ratio $= 1.15$

Then $b = 1.00$

$c = .92$

Then we have:

$$\begin{aligned} 12 &= 1.04 \times 1.00 \times .92 e \sqrt[3]{2,000} \\ 12 &= 12.06 e \\ e &= .995. \end{aligned}$$

This means practically the standard area-ratio of .40, and therefore indicates a practicable solution along these lines. We then have:

$$\begin{aligned} p &= 1.15 \times 12 = 13.8 \text{ feet} \\ N &= \frac{101.3 \times 16}{13.8 \times .82} = 143 \\ \text{Area} &= .40 \times 113 = 45.2 \text{ sq. ft.} \end{aligned}$$

(7) Instead of the routine of problem (6) let us proceed as follows:

Take $a = 1.04$

$d = 12$ feet

pitch-ratio $= 1.10$

Then pitch $= 13.2$ feet

and $c = .90$

Take also $e = 1.00$.

This leaves the slip to be determined by the factor b . We then have:

$$\begin{aligned} 12 &= 1.04 \times b \times .90 \times \sqrt[3]{2,000} \\ 12 &= 11.79 b \\ b &= 1.02. \end{aligned}$$

This indicates a slip of about .17, and we have therefore:

$$N = \frac{101.3 \times 16}{13.2 \times .83} = 148$$

$$\text{Area} = .40 \times 113 = 45.2 \text{ sq. ft.}$$

(8) Let us take next a small boat with displacement given in pounds, and speed in miles per hour:

Let $W = 18,000$.

Type, launch.

Propeller, one, 4 blades.

Speed, 12 miles per hour.

Take $a = 1.4$

slip-ratio $= .20$

pitch-ratio $= 1.2$

Then $b = .94$

$c = .95$

Take also $e = 1.00$

Then we have from (4):

$$\begin{aligned} d &= .90 \times 1.4 \times .94 \times .95 \sqrt[3]{18,000} \\ \text{or } d &= 29.5 \text{ inches} \\ p &= 1.2 \times 29.5 = 35.4 \text{ inches} = \text{say } 3 \text{ feet} \\ N &= \frac{88 \times 12}{3 \times .80} = 440. \end{aligned}$$

(9) Let $W = 8,000$.

Type, launch.

Propeller, one, 4 blades.

Speed, 18 miles per hour.

Take $a = 1.35$.

Assume d not to exceed 24 inches.

Take pitch-ratio $= 1.40$.

Then pitch $= 1.4 \times 24 = 33.6$ inches $= 2.8$ feet.

$c = 1.05$.

Also take $e = 1.00$.

This leaves slip to be determined by the factor b .

Then we have:

$$\begin{aligned} 24 &= .90 \times 1.35 \times b \times 1.05 \sqrt[3]{8,000} \\ 24 &= 25.5 b \\ b &= .941. \end{aligned}$$

From Fig. 35 this implies a slip-ratio of about .20, and we have therefore:

$$N = \frac{88 \times 18}{2.8 \times .80} = 707$$

$$\text{Area} = .40 \times 452 = 181 \text{ sq. inches.}$$

(10) Let $W = 4,800$.

Type, launch.

Propeller, one, 2-bladed, reversible.

Speed, 8 miles per hour.

Take $a = 1.55$

pitch-ratio $= 1.30$

slip-ratio $= .22$

Then $b = 1.00$

$c = .89$

Take $e = 1.15$.

Then we have:

$$\begin{aligned} d &= .9 \times 1.55 \times .89 \times 1.15 \sqrt[3]{4,800} \\ &= 24 \text{ inches} \\ p &= 1.3 \times 24 = 31.2 \text{ inches} = 2.6 \text{ feet} \\ N &= \frac{8 \times 88}{2.6 \times .78} = 347. \end{aligned}$$

The diameter may be reduced somewhat by a decrease of pitch-ratio and increase of revolutions.

Thus, with all other values the same, let us take pitch-ratio = 1.1, and hence $c = .89$. Then the diameter of the propeller will be reduced in the ratio of the two values of c or .89 to 1.00, and we have:

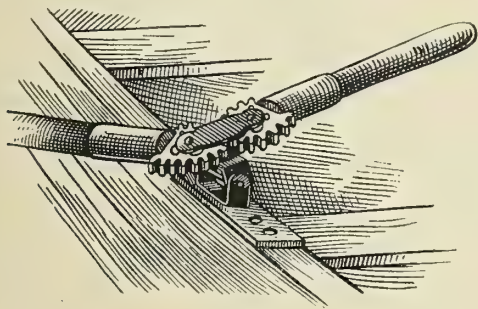
$$\begin{aligned} d &= .89 \times 24 = 21.36 \text{ inches} \\ p &= 1.1 \times 21.36 = 23.5 \text{ inches} = 1.95 \text{ feet} \\ N &= \frac{8 \times 88}{1.95 \times .78} = 463. \end{aligned}$$

Examples might, of course, be indefinitely multiplied, but the various cases here discussed will sufficiently serve to illustrate the principal problems which may arise in connection with propeller design, and the manner of their treatment by the method given.

It must not be supposed that any set of equations or printed directions can take the place of experience in propeller design, or can properly equip the amateur to confidently undertake all cases of such design with which he may come into contact. It is intended, however, to provide, by the above method, such guidance as shall best serve to direct the steps of those not familiar with the intricacies of the problem of propeller design, and at the same time to provide for those equipped with some measure of personal experience, a method of procedure simple and convenient in operation, and well adapted to the application of such individual judgment or experience as may be brought to bear on the case in hand.

Articulated Oar.

An ingenious invention, which may or may not be already known in the United States, is being talked about just now in the French journals. It is an articulated oar for row boats, which enables the rower to row his boat while facing in the direction in which he is moving instead of turning his back, as the old system requires. The inventor is M. Doyen, of 66 rue de Namur, Brussels, Belgium.



It is claimed for this invention that while permitting the rower to face in the direction in which he desires to propel the boat, it does not in any other manner affect the old system of rowing or weaken the effect of the stroke. Consequently, no training or practice is required to enable any oarsman to use the new oar. It is formed of two arms, which are joined at the oarlocks by articulating toothed sectors, which are firmly attached to the boat. This mechanism produces an inversion in the movement of the two arms of the oar. The cleats which join the two arms are so adjusted that the oarsman gives to the articulated oars the same movements of raising or lowering and of propulsion as with ordinary oars. The accompanying illustration gives an idea of the invention.—*Daily Consular Report*.

TELEPHONES ON THE GREAT NORTHERN STEAMSHIP MINNESOTA.

BY A. F. BOARDMAN.

The telephone system on this ship is said by the manufacturers to be the largest installation of the kind on shipboard. Its only equal will be the one now being installed in the sister ship, the *Dakota*, of the same company. The system here employed is of especial interest to the telephone man, but is, of course, of great interest to the marine man. There are two distinct installations: one, the general, which is a regular telephone exchange connecting the various state rooms with the steward's office. The other is a special system which connects the executive offices of the ship with each other. Both systems were furnished by the Electric Gas Lighting Company of Boston, Mass.

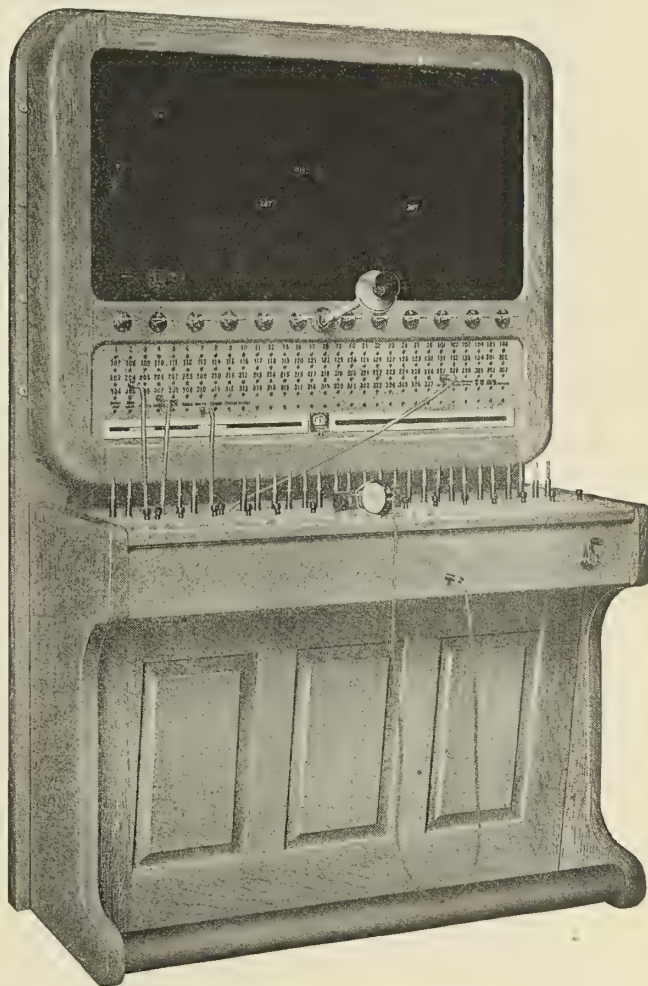


FIG. 1.—GENERAL SYSTEM SWITCHBOARD.

The switchboard of the general system is placed in a separate room in the after part of the vessel, and consists of a 150-line switchboard mounted in a light oak cabinet to match the general surroundings. Fig. 1 will give a good idea of the appearance of the switchboard. For the interest of those versed in telephony it may be stated that the system employed is what is known as a central energy system. The switchboard is a combination annunciator and telephone switchboard, which the manufacturers have given the trade name of "Annunciphone." The part of the instrument on which the signals are received is fitted with the regulation navy standard marine annunciator drop. This drop has peculiar advantages over the ordinary telephone signal in the fact that it cannot be shaken out of place by any movement of the vessel, or by any shock however severe, the electric current being the only means for displaying the signal. Below the signals are the telephone connections or "jacks." These jacks connect with each state room and by means of plugs connected by cords to the

instruments enable the operator in charge to communicate with any state room at will.

Besides the instruments in the state rooms, telephones are installed in almost every part of the vessel, such as the captain's private state room, the ladies' drawing room, the smoking room, library, kitchen, pantry, steward's department, laundry, chief engineer's office, chief electrician's office, so that no matter what a passenger's wants may be, he can be connected with the department desired. To call the steward the passenger simply needs to push the button on the face of the telephone. The call is then received on the annunciator, as in an ordinary annunciator system, but if the passenger has a verbal communication to make to the steward, or anyone else, he pushes the button and at the same time lifts the receiver from the hook and places it to his ear. Pushing the button sends an electrical impulse into the annunciator and causes the number of the room to "drop" and display itself on the face of the instrument. The operator in charge thereupon inserts one of the plugs shown into the jack corresponding



FIG. 2.—WATER-TIGHT TELEPHONE FOR EXPOSED POSITIONS.

in number to the room indicated. He is then in telephonic communication with the passenger and can answer his questions, or send a steward, or perform whatever other service is required. Should the passenger wish to communicate with some other passengers, or with some of the departments on the ship, such as the laundry or the doctor, the operator at the switchboard, by inserting one of a pair of plugs into the jack corresponding to the passenger's room, and the other plug of the same pair into that of the room desired, can enable them to talk directly with each other. If it should be required, as many as four different rooms may be connected up together in this manner, so that four people can join in the conversation. When this conversation is finished and the talking parties hang up their receivers, a target in the slot below the switchboard indicates that that particular pair have finished talking.

This switchboard is also provided with a jack for connection with outside lines, so that when in port any passenger on the ship can communicate through the switchboard with anybody on the regular city exchange on shore, or the captain of the vessel can communicate directly with the agents of the ship without leaving his cabin.

The telephone in the room, shown in Fig. 4, is a very neat looking instrument, and is said by the manufacturers to be the latest development in interior telephones. The instrument is flush except for the projecting mouthpiece. The exposed parts are of metal and finished in a dead black to match the hardware trimmings of the state room.

The navigating system is independent of the switchboard and is equipped with an entirely different sort of instrument. In this system the party making a call calls up the party desired directly by simply pushing the appropriate button and placing the receiver to his ear; he is then immediately in communication with the party desired.

The instruments on this system, where they are exposed to the weather or other detrimental conditions, are mounted in a heavy, brass, watertight case, as shown in the illustrations, Figs. 2 and 3. The instruments themselves are so made as to reduce to a minimum dangers from spray and rain, should the same enter the instrument while being operated. The use of telephones in this

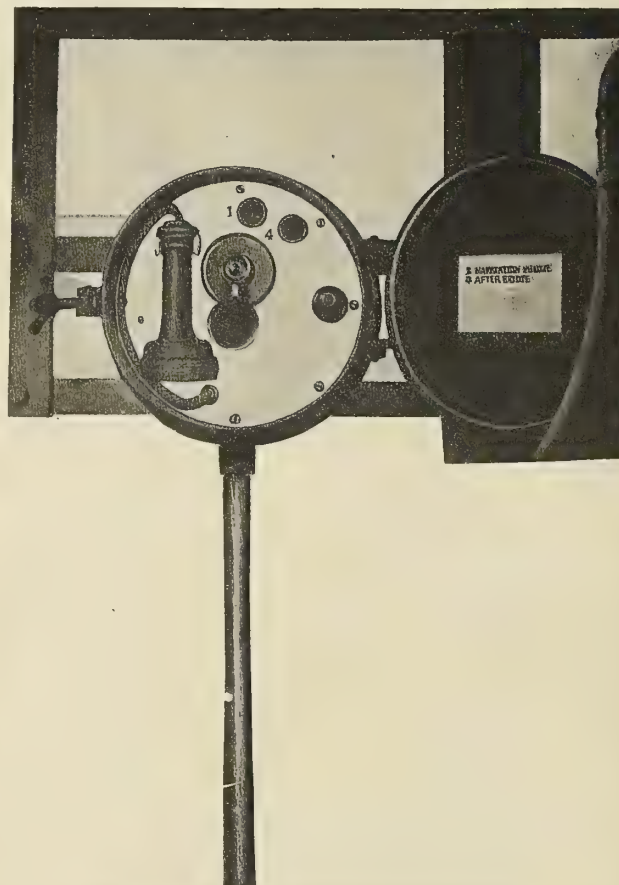


FIG. 3.—MARINE TELEPHONE ON FORWARD BRIDGE.

manner, where they form an important part of the navigating equipment of the ship, is quite an innovation, and these instruments are built in a manner never before employed in telephone construction. The exterior consists of a brass shell, to which is attached an iron conduit carrying the circuit wires. On the inside of the case is a ridge upon which a plate carrying the mechanism is fastened by means of screws. Between the plate and the ridge is placed a rubber gasket. The receiver cord is covered with a rubber tube and a rubber earpiece is provided for the receiver, on account of the noise or the rushing sound of the wind, and to keep out rain or sleet. The transmitter is made with an elbow mouthpiece arranged to prevent entrance of moisture. Another unique feature is the hook switch. The hook upon which the receiver hangs is a dummy support, the switch itself being shown to the right of the plate. It is operated by the cover, so that when the cover is closed the circuit of the telephone is open, and when the cover is open the telephone circuit is closed. An



THE GREAT NORTHERN STEAMSHIP MINNESOTA.

interesting feature is the fastening mechanism which holds the cover to the front of the box. The pitch of the thread of the thumbscrew and locking bolt is made quite coarse, so that it can be rapidly unfastened and thus save time in the case of an emergency.

On the *Minnesota* the following stations are connected by this navigating system: the forward navigating bridge, main navigating bridge, after navigating bridge, port and starboard engine room, crow's nest, pilot house, chief engineer's office, chief electrician's room, central electric lighting station and dynamo room. The instrument shown in Fig. 3 is the one situated on the forward navigating bridge. This connects only with the main and after

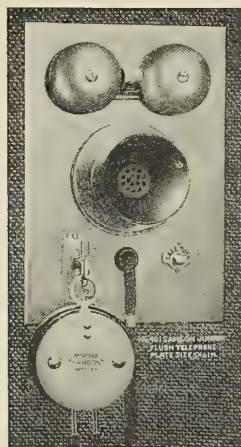


FIG. 4.—FLUSH METAL WALL TELEPHONE.

navigating bridges. One advantage of the system employed for the navigating plant is that each particular station can call only those stations which it is necessary in the course of his work for him to call. The navigating officer alone is able to talk to any of the eleven stations on this system. This can be varied to suit the requirements in any particular case.

The use of the telephone in the crow's nest is worthy of passing note. By its use the lookout is in immediate communication with the officer in charge, no matter what the weather is, and it will be understood by all that such a communication is much more sure than would be the case were he to have to call out his observations, as would otherwise be the case.

A 90,000-pound ingot, which is intended for the United States battleship *Vermont's* shafting, was recently cast at the National Steel Foundry Company, New Haven. The ingot is 50 inches in diameter and 13 feet long. To cast it the product of two 25-ton furnaces was required, and it took 35 minutes for the molten liquid to flow into the molds. Twelve other ingots of similar size are to be turned out by this company to form sections of the shafting of the battleships *New Hampshire* and *Rhode Island*.

POWERING SHIPS.

BY EDWIN CERIO, NAVAL ARCHITECT.

PART III.

V. ADDITIONAL RESISTANCES.

Usually when estimating horsepower only skin and wave resistance are considered; all residual resistances are implicitly comprehended in the latter, as in practice under wave resistance is understood the residual part of the total resistance after deducting that part due to friction. But in special cases, and whenever great accuracy is required, it is advisable to study separately the single additional resistances, the principal of which will be here analyzed.

(a) *Resistance Due to Irregularity of the Ship's Motion.* This depends upon the changes in speed and direction undergone by the ship in her progress. Propellers, as they are constructed up to date, are not able to secure a uniform motion, and this is the principal cause to which the additional resistance above mentioned must be ascribed.

(b) *Resistance Due to Stormy Sea.* This is also of such a nature that, like the preceding, it cannot be estimated *a priori* when designing a ship. Speed trials are in ordinary circumstances made in fair conditions of wind and sea; therefore stormy weather may be looked upon as an accidental cause of resistance not to be taken into account when powering the ship's engines. According to Durand, the influence of a stormy sea on resistance is less felt in a vessel when she has a considerable length, sufficient freeboard, is steady, and has large dimensions; consequently great weight.

(c) *Air Resistance.* In the first steamships this cause of resistance was negligible as compared to the importance it had in sailing vessels, but in modern ocean liners air resistance comes to be a considerable factor in the problem of powering ships, owing to the huge superstructures so common now in such craft. For a rough calculation the resistance of air may be taken into account as a percentage of the total resistance, but a more careful method is that of estimating it by means of formulæ. Calling W_a the air resistance and A the projected area of all exposed planes on a plane normal to the direction of motion, the general expression of this additional resistance is:

$$W_a = kA v^2.$$

This equation may be written in a more general way:

$$W_a = kA (V \mp V_1 \cos \alpha)^2.$$

W_a is given in pounds,

A in square feet,

V = speed of the ship, in knots,

V_1 = speed of the wind (if any), in knots,

$k = 0.005$.

The signs — and + to be employed according to the direction in which the wind blows. Air resistance varies with the square

of the speed, while the principal resistances of a vessel in motion vary according to a higher power of the speed; therefore for great speeds this additional resistance due to air is of little importance as compared to skin and wave resistance. It assumes considerable values only when a strong wind is blowing against the ship, in which case it may sometimes as much as double the ordinary resistance. The formula given above is very useful to make corrections of the results obtained at trial trips carried out in stormy weather.

(d) *Resistance Due to Bilge Keels.* Bilge keels, and all other structures which in some points modify the ship-shaped forms of a vessel's hull, are apt to produce considerable changes in resistance; it is possible to appreciate the influence of bilge-keel resistance only by means of model experiments of ships similar to those which are to be designed. Very important researches on the influence of bilge keels were made at Spezia by Signor Rota of the Italian navy. A model of a battleship and one of a torpedo-boat destroyer were employed in a series of most important ex-

periments, the results of which furnish the most reliable and exhaustive source of information on the subject. Table IX gives the dimensions of the vessels in full size:

TABLE X.—BILGE KEELS ON THE BATTLESHIP SARDEGNA.

Dimensions of Bilge Keels.	1	2	3	4	5	6
Length.....	164'	197'	164'	197'	164'	197'
Breadth.....	3'-3 ¹¹ / ₁₆ "	3'-3 ¹¹ / ₁₆ "	3'-3 ¹¹ / ₁₆ "	3'-3 ¹¹ / ₁₆ "	3'-3 ¹¹ / ₁₆ "	3'-3 ¹¹ / ₁₆ "
Height.....	1'	1'	1'	1'	1'	1'

The reduction of speed due to bilge keels, when employing the

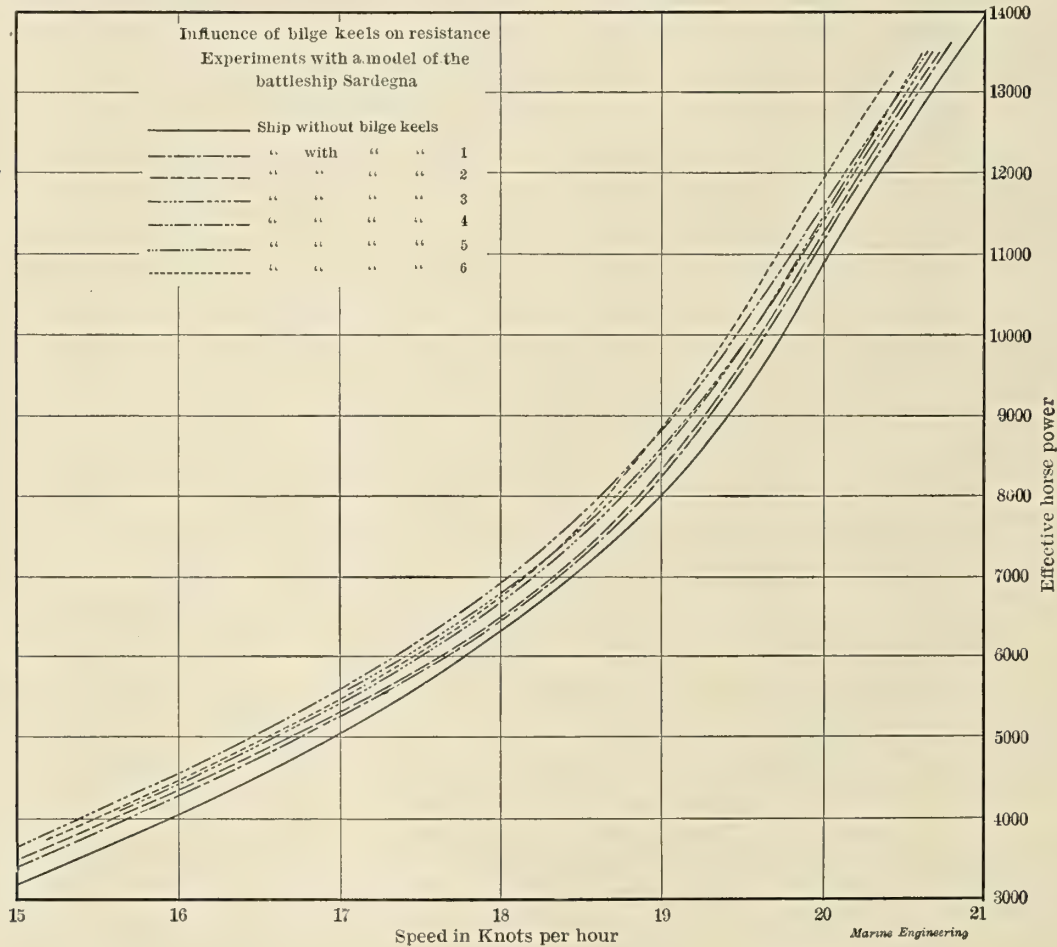


FIG. 6.

periments, the results of which furnish the most reliable and exhaustive source of information on the subject. Table IX gives the dimensions of the vessels in full size:

TABLE IX.

Principal Data.	Battleship Sardegna.	32-Knot Torpedo Boat.
Length between perpendiculars..... Ft.	410	229
Extreme breadth..... "	77.5	22.7
Mean draft..... "	28.6	5.05
Wetted surface (without bilge keels)..... Square ft.	33,640	4,400
Displacement..... Tons.	13,700	320

same horsepower, is shown in Table XI at the actual speed of the vessels without bilge keels.

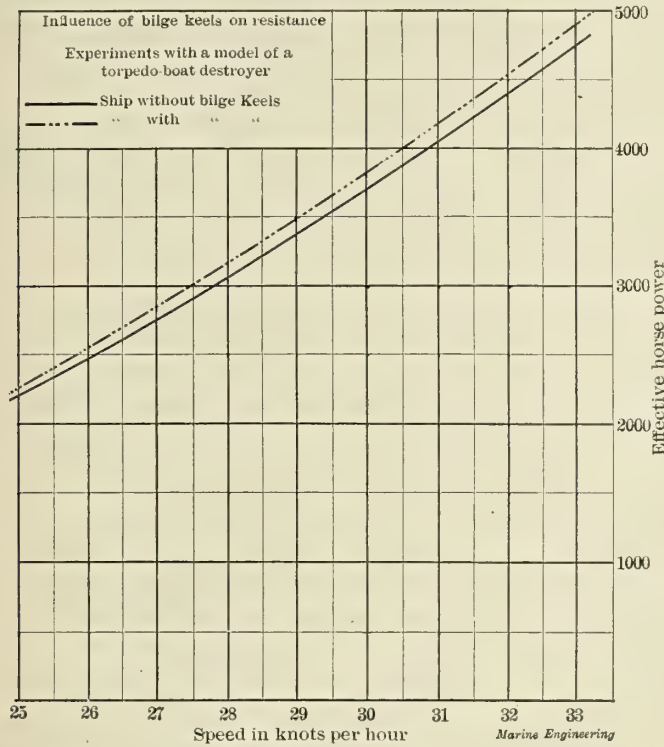
TABLE XI.—REDUCTION OF SPEED DUE TO ADDITIONAL RESISTANCE OF BILGE KEELS.

Ship.	Speed in Knots per Hour.					
	Without Bilge Keels.	With Bilge Keels.				
		No. 1	No. 2	No. 3	No. 4	No. 5
Sardegna.....	20	19.92	19.88	19.84	19.77	19.82
Torpedo-boat destroyer..	32	31.62				

The accompanying diagrams (Figs. 6 and 7) show the increase

When designing a ship it will be always safe to consider an increase of at least 5 percent to the resistance if bilge keels are to be adopted.

(e) *Resistance Due to Shallow Waters and Limited Transverse Section of the Liquid Mass in which the Vessel Moves.* A characteristic of modern researches is the attention given to the influence of a limited depth of water on resistance; the navigation of a vessel in shallow waters, in a canal, or in the proximity of shoals and banks, produces a disturbance in the stream lines which are compelled to pass through a section of limited dimensions. Thus modified, the stream lines require an additional amount of energy to pass from the vessel to the water, and consequently an additional resistance is originated.



A remarkable example of the influence of shallow waters may be observed in the values of the Admiralty coefficient for the following warships at a speed of 20 knots:

	I. H. P.	C.
Gunboat <i>Sharpshooter</i>	3,500	186
Third-Class Cruiser <i>Medusa</i>	10,000	159
Second " " <i>Terpsichore</i>	9,000	198
First " " <i>Edgar</i>	11,000	276
" " " <i>Blenheim</i>	12,500	278

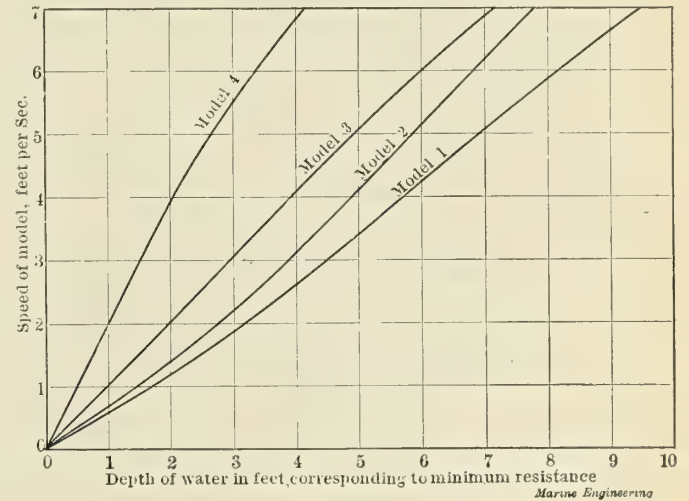
Except for the *Medusa*, the trials of these ships took place in very deep water; the *Medusa* trials giving such a low value of C were carried out in Stokes Bay, in which, as is well known, the depth of water is very limited, and where no more trials are now conducted by the British Admiralty.

Another instance illustrating the same fact is that of the U. S. S. *New York*, cited by Durand; in this vessel, when passing on a point where the depth of water was 37 fathoms, it was observed that the revolutions fell off and the steam pressure increased. Taylor has given the limits of depth at which the influence of the bottom produces no change of resistance, basing these limits only on the speed of the vessel. The results of Taylor's investigations on the distribution of steam lines have been obtained without taking into account the size of the vessel, and therefore the conclusions arrived at have no practical value. The question has been in the last few years thoroughly studied by Colonel Rota of the Italian navy.*

The diagrams contained in Fig. 8 have been worked out with the results obtained by model experiments carried out at Spezia, and show the depths required in order that the bottom may have no influence on the resistance. The models experimented with had the dimensions given in Table XII.

Dimensions.	Model No.			
	I	2	3	4
Length..... Feet.	14.03	12.24	12.24	14.37
Breadth..... "	2.59	2.13	1.87	1.52
Mean draft..... "	0.92	0.68	0.68	0.53
Displacement..... Pounds.	1069.2	555.5	488.3	354.1
Block coefficient.....	0.51	0.50	0.50	0.49

The results of the model experiments may be extended to full-size ships bearing any ratio of similitude to the models; Table



XIII is a rough application of such results to ships, varying from 12,000 tons to 1,000 tons in displacement, the limits of depth at which the influence of the bottom is no longer felt being given for speeds varying between 12 and 26 knots.

Ship.	Corresponding to Model No.				
	I	2	3	4	4
Displacement..... Tons.	12,000	8,000	6,000	3,000	1,000
Length..... Feet.	408	385	361	380	263
Breadth..... "	75.5	67	55	40.3	28
Mean draft..... "	26.6	21.4	20.2	13.8	9.6
Minimum Depth in Feet for no Change in Resistance.					
Speed in Knots.	12.....	138	108	89	75
14.....	177	158	125	115	92
16.....	200	177	145	125	118
18.....	223	197	161	141	125
20.....	246	214	181	161	145
22.....	270	233	204	184	164
24.....	...	253	230	204	184
26.....	...	...	...	227	204

A ship navigating in a mass of water having a narrow transverse section, as is the case in the navigation of ships in canals

experiments, has kindly put at my disposal the material of observations contained in his "*Vasca, etc.*" and other works in which the problem of resistance is investigated, and I wish to express to him my thanks for all the valuable information obtained through his kindness.

*Colonel Rota, the eminent authority in all matters connected with model

and rivers, experiences the influence of the disturbance in the stream lines, and the amount of power necessary for propulsion is much greater than that which she would require when navigating in a mass of water of practically indefinite section, as is the case of the open sea. Many experimental researches have been made to determine the resistance of ships in a canal; those conducted by Elnathan Sweet have led to the formula*

$$R = \frac{0.10303 \text{ } sv^2}{r - 0.597},$$

where R = resistance in pounds,
 s = wetted surface in square feet,
 v = speed in feet per second,
 r = ratio of section of canal to that of vessel.

Calling W' the resistance of a ship in a channel and W the resistance of the same in a mass of water of indefinite section, for speeds varying between 1 1-2 feet and 3 feet per second:

$$W' = \text{from 2 to 3 } W.$$

If a is the area of the transverse section of the vessel and A that of the channel, the ratio $\frac{W'}{W}$ grows with the speed and is indirectly proportional to the ratio $\frac{A}{a}$. From the experiments of de Mas† some conclusions of great importance may be drawn, such as those contained in Table XIV.

TABLE XIV.

$\frac{A}{a}$.	Values of $\frac{W'}{W}$ at a speed of :	
	1 Foot per Second.	4 Feet per Second.
8	1.00	1.20
5	1.40	2.20
3	2.60	5.80

Experiments by Rota lead me to conclude that the law of dynamical similitude cannot be applied to vessels navigating in canals as it is to those moving in water of unlimited section; this is evidently due to the fact that in the case of a channel, besides the resistance due to the skin-friction of the ship, the resistance generated by the friction of the banks and bottom of the channel ought also to be considered. This is not possible in practice, owing to the meager amount of data obtainable. In Fig. 9 a diagram is given showing the curves of resistance of the model of the river boat *Jeanne*‡ at different speeds and for different sections of canals.

Experiments were made to determine the influence of depth of water on the resistance of torpedo boats first by Capt. A. Ras-mussen,** and lately in the German navy with a Schichau boat, the *S 119*, at Neukrug, near Danzig; the Baltic, in that part called the Frische Haff, where Neukrug is situated, presents remarkably favorable conditions for such trials as those carried out by Herr Marine Baumeister Paulus, who communicated the results to the Schleswig-Holsteinischer Bezirksverein Deutscher Ingenieure. As these results constitute a source of valuable and reliable information, we present the principal data obtained in these trials.

*Durand, "Resistance and Propulsion of Ships."
†Recherches expérimentales sur le matériel de la batellerie, par T. B. de Mas, Paris, 1891-97.
‡The reason the model of this ship was chosen is that the full-size vessel has already been the object of important experiments carried out by de Mas, the results of which are cited by Durand.
**Trans. of the Inst. of Nav. Arch., 1899. See also MARINE ENGINEERING, November, 1902, page 581; and April, 1903, page 202.

The *S 119* is the last boat built of a series of destroyers completed in 1903; she has the following characteristics:

Length at the water-line.....	207.60 ft.
Extreme breadth	22.96 ft.
Draft	8.86 ft.
Displacement	345 tons
Speed	30 knots
Indicated Horsepower	6,500

The trials in question took place in September, 1903, under uncommonly fair conditions of weather, and lasted a week; the measured mile was run 84 times at different depths of water, varying from a depth of 23 feet to one of 200 feet, beginning with the *minimum* depth and ending with the maximum, and *vice versa*

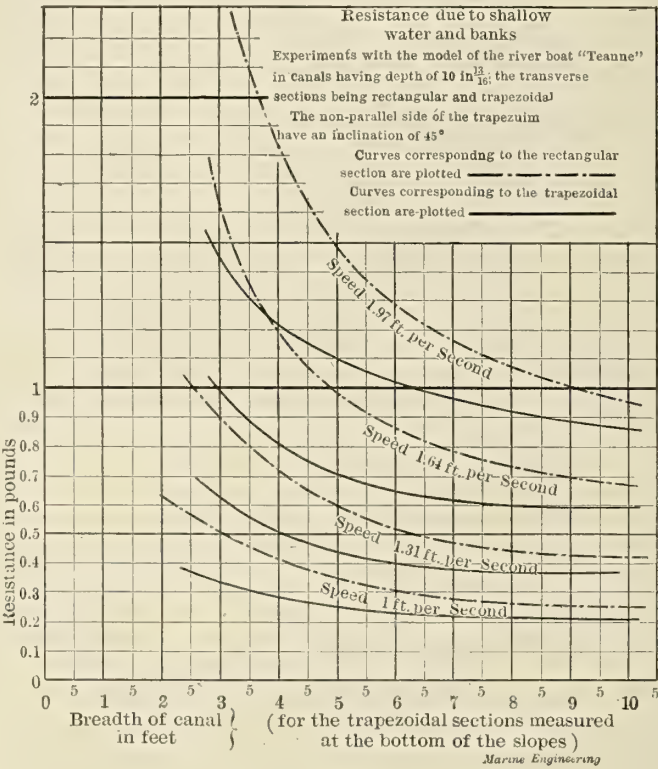


FIG. 9.

in the next series of runs. During the trials the speed, indicated horsepower, number of revolutions, trim, and wave configurations were measured with great exactness. Tables XV, XVI, and

TABLE XV.

IHP.	Speed at a Depth of :					
	200 Ft.	130 Ft.	82 Ft.	49 Ft.	34 Ft.	23 Ft.
5,680.....	27.17	26.93	26.55	27.20	27.66	27.82
4,600.....	25.13	24.86	24.56	23.58	25.50	25.95
4,000.....	24.01	23.73	23.46	21.80	23.52	24.52
2,700.....	21.48	21.28	21.09	20.00	17.75	15.84
2,000.....	20.00	19.84	19.72	19.08	16.94	15.11
1,500.....	18.75	18.67	18.58	18.20	16.68	14.84
1,000.....	17.16	17.16	17.06	16.86	16.22	14.44
500.....	12.2	12.2	12.2	12.2	12.2	12.2

TABLE XVI.

Speed.	IHP at a Depth of :					
	200 Ft.	130 Ft.	82 Ft.	49 Ft.	34 Ft.	23 Ft.
27 Knots.....	5,590	5,715	5,920	5,605	5,315	5,195
24 ".....	3,995	4,140	4,290	4,710	4,110	3,870
21 ".....	2,465	2,560	2,650	3,550	3,525	3,510
18 ".....	1,240	1,250	1,285	1,410	2,815	3,210
15 ".....	600	600	600	615	640	1,800
12 ".....	285	285	285	285	285	285

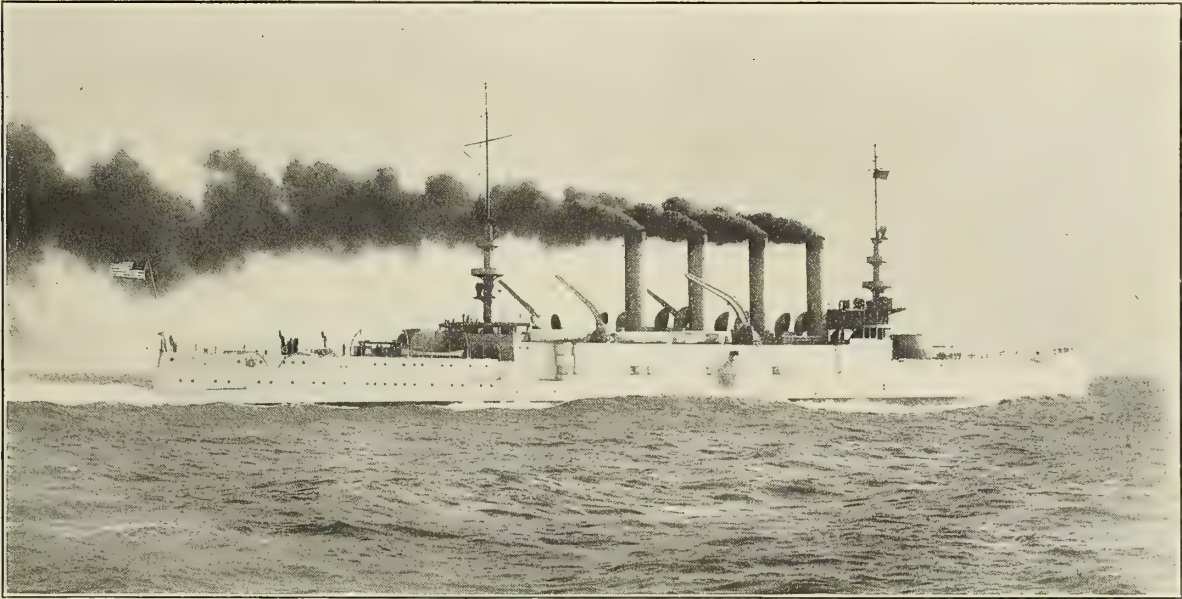
TABLE XVII.

Number of Revolutions.	Speed at a Depth of :					
	200 Ft.	130 Ft.	82 Ft.	49 Ft.	34 Ft.	23 Ft.
270.....	26.75	26.55	26.22	26.57	27.15	27.31
250.....	24.84	24.66	24.34	23.60	25.03	25.32
200.....	20.52	20.36	20.42	19.56	17.55	16.05
150.....	16.66	16.65	16.65	16.52	16.14	14.55
100.....	11.7	11.7	11.7	11.7	11.7	11.7

XVII contain the values of speeds for constant values of indicated horsepower, those of the indicated horsepower for constant values of the speed and the values of the speed for constant values of the number of revolutions, all these values being given for the different depths at which the experiments were carried out. Observing the results contained in the above tables, several important conclusions may be drawn, which apply to torpedo-boats in general and in particular to all ships bearing similitude to the one under experiment, necessary allowance being made for the difference in size, etc.

THE TRIAL TRIP OF THE ARMORED CRUISER MARYLAND.

On the 27th of January the new armored cruiser *Maryland*, under construction by the Newport News Shipbuilding and Dry Dock Company, was given a trial trip on a course extending for a distance of 44 knots from Thatchers Island off Gloucester to Cape Porpoise. The mean speed, without tidal correction, was figured out as 22.306 knots, and the tidal correction as computed added to this figure one-tenth of a knot, making the corrected speed 22.406. The indicated horsepower of the main engines is given as 27,571 and of the main engine and all auxiliaries 28,475. The coal consumed per square foot of grate figured out at 50.72 pounds per hour; that consumed per indicated horsepower of main engines was 2.872 pounds per hour, and the amount per indicated horsepower of all machinery 2.816 pounds. The air pressure in the fire-room was represented by 1.93 inches of water, the steam pressure being 300 pounds per square inch. This pressure was reduced at the engines to 250 pounds by means of special reducing valves, for the purpose of drying the steam and to a slight degree superheating it on its entrance into the high-pressure cylinder. The mean effective pressure reduced to the low-pressure cylinder was



THE UNITED STATES ARMORED CRUISER MARYLAND ON TRIAL.

At a depth of 130 feet the influence of the bottom is not felt by torpedo-boats; at a speed of 27 knots the most unfavorable depth is that of 82 feet, as at this depth the decrease in speed is nearly 21-2 percent. At a speed of 12 knots, navigating at different depths of water, no increase in the indicated horsepower is noticeable.

An instance proving the usefulness of comparison in problems of the kind of those here examined, is furnished by the Russian cruiser *Askold*. This vessel, built by Krupp at the Germania yard, Kiel, underwent her first trials in the Eckenfoerde Bay, under unfavorable conditions of depths of water; basing the comparison on Rota's model experiments, it was calculated that in deep water one-half mile would be gained in speed. This assumption was proved true by trials carried out at sea under the same conditions as those obtained in the Eckenfoerde trials.

There are still some other causes which increase the resistance of ships: for example, the slope of currents, etc.; but these are not of such importance as to be considered when powering ships, and therefore are not mentioned in this study.

(To be concluded.)

51.76 for the starboard side and 52.35 for the port, giving a final mean of 52.055 pounds per square inch. The engines developed 128.5 revolutions per minute as the mean for the four-hour trial.

The *Maryland* is a steel armored cruiser with a displacement of 13,680 tons, having a length on the load water-line of 502 feet, a beam of 69 feet 6 1-2 inches, and a normal draft of 24 feet 1 inch. She was designed for a speed of 22 knots, which was to be developed by engines giving an indicated horsepower of 23,000 at 120 revolutions per minute. The coal supply is represented by 900 tons at the normal displacement, increased on full load to 2,024 tons. The full complement of officers and men numbers 41 and 787 respectively.

The twin-screw engines are placed in separate water-tight compartments and have each four cylinders measuring respectively 38 1-2 inches, 63 1-2 inches, and two of 74 inches diameter, with a common stroke of 48 inches. Steam is supplied by sixteen water-tube boilers of the Babcock and Wilcox marine type, placed in closed separate water-tight compartments. The total grate surface of the boilers is 1,600 square feet, and the heating surface

70,944 square feet; thus giving a ratio of heating surface to grate area of 44.34 to 1.

The main battery consists of four 8-inch breech-loading rifles mounted in pairs in turrets on the center line, forward and aft, and protected by 6 to 2 inches of armor; fourteen 6-inch rapid-fire guns mounted on the broadsides; eighteen 3-inch and twelve 3-pounder rapid-fire guns; besides eight 1-pounder automatic guns, six .30 automatic guns, two machine guns, two 3-inch field guns, and two 18-inch submerged torpedo tubes. The armor belt has a width of 9 feet, of which 5 feet is below and 4 feet above the normal load water-line. This belt is 6 inches in thickness at the top and tapers to 5 inches at the bottom. It further tapers at the stem and stern to a thickness of 3 to 2 inches.

Trial Trip of the Chattanooga.

The trial trip of the cruiser *Chattanooga*, which was mentioned in the February number of MARINE ENGINEERING, was carried out under somewhat different conditions than those which have obtained in the trials of nearly all of the ships in the United States navy. There were three sets of trials, consisting of standardization trials, a four-hour official speed trial, and a twenty-four hour endurance run at two-thirds power. In the first series, which took place on the 31st of December, the ship was run over the measured mile in Newport harbor for the purpose of determining the speed at different rates of revolution of the propellers. There were fifteen of these runs at revolutions varying from 60 to 180 per minute. In order to eliminate the effect of the tide, two runs were made at each speed, one in each direction.

The four-hour official speed trial was run on the 1st of January, and dependence was placed on the standardization trials for a record of speed. This method obviated the necessity of having a standard course marked by several stake boats, and also the necessity of keeping the ship in a perfectly straight course without reference to the movements of other vessels. The trial was reported to be a great success in every way, as the engines ran evenly and smoothly, and the boilers steamed freely, maintaining without difficulty the constant pressure of 265 pounds per square inch. The smoke is stated to have been very slight in quantity and very light in color. The principal results of the trial were:

Revolutions per minute of main engine.....	180.5
Speed of ship in knots.....	16.605
Indicated horsepower.....	5,390
Coal burned per hour per square foot of grate surface.....	35.4
Indicated horsepower per square foot of grate surface.....	18
Coal burned per hour per indicated horsepower, pounds.....	1.97

This coal consumption at full speed figures out at 4.74 tons per hour, or 113.7 tons per day, giving a steaming radius on the normal coal supply of 470 tons of 4 days and 3 hours, corresponding with a distance of 1,646 knots.

The twenty-four hour endurance run started on Monday the 2d of January in the midst of a terrific gale; after running for several hours in the teeth of this gale it was decided by the trial board that the capabilities of the ship and her machinery had been thoroughly proven, and the trial was discontinued.

Examinations for inspectors and assistant inspectors of hulls are announced for March 15 and 16, by the United States Civil Service Commission. The vacancies to be filled at once are one at Grand Haven, Mich., at \$2,000; Cincinnati, Ohio, \$1,200; Seattle, Wash., \$1,500. The examinations are open to all American citizens between the ages of 25 and 55. Candidates should apply to the Commission, at Washington, for Forms 1087 and 1405.

A steam mud barge is in use on the Loire river, France. At present the river is at an extremely low level and the incoming tides deposit sand and mud. During the ebb tide the barge, which is fitted with a rake 36 feet long and weighing about 2 3/4 tons, is moved up and down the stream, dragging the bottom and loosening the silt, which is carried away by the tide.

EPOCHS IN MARINE ENGINEERING.*

BY GEORGE W. MELVILLE, REAR-ADMIRAL, U. S. NAVY.

To attempt to cover the history of marine engineering in a lecture which ought not to last much longer than an hour, would result in little more than a mere chronology and could not possibly be interesting. It seems much better, therefore, to give some consideration to the various periods or epochs in the history, considered with reference to the special inventions or changes which have characterized them. Many of these changes are clear examples of evolution, and others are instances of the adaptation of land practice to marine use, but they all serve to show the constant progress which has taken place.

THE PADDLE WHEEL.

The first marine engineering in the modern sense was the adaptation of the steam engine as already in familiar use on shore to a modification of the centuries-old method of mercantile propulsion, the oar. I believe some attempts were actually made to adapt the steam engine to a series of oars, which would have meant something like a mechanical trireme; but of course the trained mechanical sense soon saw that the collection of the oars in a revolving wheel was the correct solution. As oars had been used on both sides, so it was natural at first that the paddle wheels should be on both sides; a center wheel was also tried, but it is interesting to remark that practically about the same time that the sidewheels were used on the seaboard, the sternwheel was introduced on the western rivers. The first marine engine was the shore engine, modified to suit the circumstances; and thus on the seaboard the engine was designed and worked with what we now consider an exceedingly low pressure. On the western rivers, where the change had been made in the location of the wheel, there was also the additional change of dispensing with the condenser and using very much higher pressures. It was doubtless due to this fact—that the first non-condensing engines really carried a very high pressure—that the term "high-pressure" in the early days meant non-condensing. The reason for the difference is of course very clear; the western rivers are very shallow and it was necessary to make the machinery as light as possible; on the seaboard and the rivers of that section there was deep water, and the vessels had displacement enough to permit of heavy machinery.

Ordinarily the history of this olden time could have only an antiquarian interest for us, but we are unusually fortunate in still having with us in the active practice of his profession an engineer who saw the first commercially successfully steamboat, the *Clermont*, so that through our "grand old man of engineering" Mr. Charles H. Haswell, we still have a living connection with that earlier time. One of my former assistants some years ago contributed a series of articles to one of the engineering magazines and has given some data as to the performance of the *Fulton*, the first American steam man-of-war, whose machinery was designed by Mr. Haswell, who was also her first chief engineer. An extract from the *Fulton's* steam log for January, 1838, shows that the maximum steam pressure was eleven pounds, the vacuum twenty-four inches, and the maximum revolutions per minute eighteen.

THE SCREW PROPELLER.

As time passed on and experience was gained, there was naturally an improvement in workmanship and design and a moderate increase in steam pressure, but about 1836 the screw propeller was brought forward for driving large vessels. This was not the first application of the screw propeller, which had been used successfully on a small steamer or launch about 68 feet long as early as 1804 by Col. John Stevens, the grandfather of Col. Edwin A. Stevens, who is now so active in marine work, and the father of

*Address delivered before the American Society of Mechanical Engineers, January 31, 1905.

Robert L. Stevens, who was the most famous of the name for his work in connection with marine engineering. It is to be noted that this date is prior to that of the building of the *Fulton*, but naturally, in the first steam war vessel, it was not considered desirable to do anything of an experimental nature, a condition which has obtained to some extent ever since, and probably always will. It was about ten years later that the propeller began to come into general use, and entirely displaced the paddle wheel for ocean-going steamers. The reason for this change is easy to see on a little consideration. On a long ocean voyage the change in displacement is due almost entirely to the consumption of fuel. In the case of the propeller this makes practically no difference in its immersion or in its efficiency, while in the case of the paddle wheel the immersion of the floats would be changed, with a diminution of efficiency. Further than this, the paddle boxes offered very great resistance to strong head winds and also brought severe stresses on the ship, due to rolling in heavy seas; the propeller not being affected by either of these last two causes. To-day for work in deep water, the screw, of course, is the only propeller; but for river work, particularly in shallow rivers, the paddle wheel is still used. Efforts have been made, and some of them exceedingly ingenious, to adapt the propeller to use in shallow water, and a certain amount of success has attended the efforts of such brilliant engineers as Yarrow and Thornycroft, not to mention our own Mr. Charles Ward, who has done some work in this connection. The fact remains, however, that damage by sand bars, snags, etc., may easily render screw machinery inoperative, while, as expressed by an engineer who had designed many western river boats, "any wood butcher can repair a stern wheel."

Probably fully as much has been done to improve the design of propellers, as time has gone on, as any other part of marine machinery. In the early days the rules for propeller design were exceedingly crude, but with the slow engine-speeds which then obtained the effects were not so noticeable. As engine-speeds increased, however, it was seen that these old rules were utterly inadmissible. There is no good excuse, however, for progress having been so long delayed; for the designs still remained too crude even after Isherwood's famous Mare Island experiments in 1868. Probably one of the great troubles with screw-propeller design at the beginning was the mistake made in considering the action of the screw as analogous to that of a bolt working in a nut, from which it was inferred that the smaller the slip the greater the efficiency. As a matter of fact, a screw propeller is really a pump for driving a mass of water astern, the reaction from which drives the vessel ahead. When this had once been realized, it was seen that there must be a certain amount of slip, and that under proper conditions there could be relatively large slip and still high efficiency.

Multiple screws were used as early as our Civil War on some vessels known as "tin-clads" on the Mississippi, their adoption being necessitated by the shallow draught. Twin screws were first used in war vessels, where the necessity for keeping the machinery below the deck would not allow of all the power being conveniently used on a single shaft, but the great advantage they possess of security against total disablement and for maneuvering soon made them the rule for all naval vessels large enough to admit of them. They were much longer in coming into the merchant service, where the limitations on naval machinery do not obtain; but since the era of the very large transatlantic steamers—beginning with the *Paris* and *New York* and the *Teutonic* and *Majestic*—all very large vessels have been built with twin screws.

In the navies of France and Germany, the triple screws have been used to a considerable extent, and I used them myself in our own fast cruisers *Columbia* and *Minneapolis*. My own belief is that they have decided merit for vessels using above 10,000 horsepower. This view, however, was not shared by my board colleagues in the navy department, so that their marked success in the two vessels named was not allowed to be repeated in later designs.

With the advent of the steam turbine as a prime mover in ships multiple screws have again come to the front, this time on account of the extremely high speed of rotation of the shafts. The *Turbinia*, the first vessel of this kind, had three shafts with three propellers on each; and the destroyers *Viper* and *Cobra* had four shafts with two propellers on each. The merchant vessels, beginning with the *King Edward*, have been fitted with three shafts.

Before leaving the propeller, I may mention, in connection with the improvements in its design, the care that is now taken to avoid needless friction, by making the hub spherical with a conical tail piece, and by putting covering plates over the bolts securing the blades to the hub so as to continue the outline of the sphere or conoid.

THE SURFACE CONDENSER.

Up to about 1860 the jet condenser was the one usually employed on board ship, which meant, of course, that the boilers were constantly fed with salt water; and this, in turn, meant the deposition of great quantities of sulphate of lime scale on the heating surface. With the low pressures then prevalent this did not greatly affect the economy of the boilers, except that, as "blowing off" to keep the density of the water down was a continuous practice, there was a certain loss of heat, and of course there was the necessity for frequent scaling of the heating surfaces. However, they were effectually protected against corrosion. About 1860 the use of surface condensation became general, and as this greatly reduced the amount of scale formed, it was practicable and safe to increase steam pressures, which accordingly resulted with a consequent reduction in the weight of machinery per unit of power.

An accompaniment of the introduction of surface condensation, which was at first supposed to be a result of it but which as a matter of fact was not, was a tremendous increase in the corrosion of the boilers and shortening of their life. This was especially noticeable in the tubes which, as the thinnest part of the boilers, gave out first. All sorts of theories were advanced to account for it, some of which we can now see to have been utterly ridiculous. Probably one of the most fanciful was that which regarded the boiler and condenser as forming a gigantic galvanic battery, the copper condenser tubes forming one pole and the boiler the other. The real facts were developed as a result of the investigation by the Admiralty Committee on boilers in the '70's, which showed that boiler corrosion was simply rusting, and had been due to gross but unintentional neglect. It had been a very common practice, particularly in naval boilers, when they were not in use, to blow out the water, and take off the manhole plates "to let them air." It was this "airing" which caused the corrosion. Now, when boilers are laid up, they are filled with water which is made slightly alkaline, and this effectually prevents corrosion.

THE CYLINDRICAL BOILER.

The early boilers in sea-going vessels were of what has been called the "box" type; that is, the boiler was a cubical box with a thin shell, the real strength being given by braces running in three directions. When surface condensation had made higher pressures possible, it was soon found that the multiplicity of braces, as pressures were increased, made an impossible condition of affairs, and this led to the design of the cylindrical boiler, whose shell was self-bracing, and left the only braces, those needed for the heads and flat surfaces. This boiler so thoroughly met the conditions arising that it has remained the favorite even up to the present day. At one time an effort was made to save room by making the boiler elliptical, but this was soon found to be unsatisfactory and impracticable, and was abandoned after only a few examples.

The earliest cylindrical boilers were single-ended, with two furnaces, but with the advent of reliable mild steel the diameters were increased and the boiler was made double-ended, with upper ends rounded to save bracing; so that the largest cylindrical

boilers to-day have as many as eight furnaces, four in each end in pairs; that is, the two furnaces at each end and on the same side of a vertical diameter have a common combustion chamber. The saving in weight due to the double-ended boiler is evident at once, and also the reduction in the feeding apparatus required.

Notwithstanding the advent of the water-tube boiler, which will be mentioned further on, and its practical preemption of naval practice, the cylindrical boiler still remains the favorite for the merchant service and has been used for pressures as high as 220 pounds, even in the largest sizes, on such vessels as the *Kaiser Wilhelm II*. The highest recorded pressure is 255 pounds, on the *Inchdune*.

THE COMPOUND ENGINE.

From a very early period steam had been used expansively in marine engines, and indeed sometimes to a ridiculous extent. Some engineers as late as the Civil War hardly seemed to realize that there was any limit to expansion, although Isherwood's famous experiments on the *Michigan* in 1861 had demonstrated conclusively that, with low pressures, only a very moderate expansion is permissible, beyond which any further expansion is attended by an economic loss. As pressures increased it was natural and correct that a higher range of expansion should be used, and this made practicable the compound engine, where the expansion occurs in two stages, the high-pressure steam from the boiler being limited to a small cylinder from which, in turn, the steam of lower pressure is exhausted to a larger cylinder. As you all know, the compound engine was invented almost as early as Watt's separate condenser, Hornblower's patent dating back to 1771, and Wolff's patent for a two-cylinder engine dating back to 1804. With the low pressures prevalent at that time the compound engine was actually at a disadvantage compared with the simple one. When pressures had reached about 60 pounds, however, the compound engine began to assert itself, the pioneer in that respect being John Elder, of the firm of Randolph & Elder, which is now known as the Fairfield Engine Works. It is interesting to note that the Allan Line of steamers, which is now the pioneer in introducing the steam turbine for an ocean-going steamer, made the last scientific stand against the compound engine, going so far as to take duplicate vessels, and engine one with compound and the other with simple engines of the same power. The actual experience with these two vessels, where the simple engine with the high ratio of expansion was constantly in trouble from breaking down, was a convincing proof that high ratios of expansion in a single cylinder were impracticable.

With the improved workmanship which had come by this time, and with the improvement in materials, to which we shall refer in a moment, which came later, the compound engine advanced to a high state of perfection, and for large powers the three-cylinder type, with one high-pressure and two low-pressure cylinders, became a favorite for all large vessels. These engines were probably as fine specimens of marine engine designing as have ever been seen, and included some exceedingly ingenious valve gears designed to secure variable expansion and an equalization of work among the various cylinders. As we shall see later, the further advance relegated these beautiful mechanisms to the engineering museum.

THE ADVENT OF MILD STEEL.

It is probably difficult for the young men in our technical schools of to-day, who are familiar almost entirely with mild steel and very little with wrought iron, to realize what a change came in engineering when the production of mild steel became a commercially reliable matter. When we look back at the way in which some of the vital elements of a big marine engine were made, we are almost inclined to wonder that the material was reliable at all. The difference between a large wrought iron shaft such as old Hughey Dougherty used to make at the Morgan Iron Works, and one of the mild steel shafts made at Bethlehem, is as great as could be imagined. Nearly the same is true of boiler plates. The young engineer of to-day would hardly know what

was meant by a lamination or a "cold shut." The very method of manufacture made it necessary to use a large factor of safety in designing, with the result that the working stresses permissible were very low and the weight of machinery inordinately high. With the advent of mild steel and the introduction of careful and systematic testing, the designer had a material on which he could place absolute reliance, so that the factor of safety could be greatly reduced. As a matter of fact the factor of safety has been reduced from 8 or 10 to 5, and sometimes as low as 4.5.

In looking back over my own experience, I do not see how we could possibly have built engines of the size and power now common with wrought iron for pistons and connecting rods and shafting, and it is, of course, absolutely certain that we could not have built the cylindrical boilers of to-day. The change began in the later '70's and had become almost complete by the middle of the '80's.

We must not fail to notice in the change to steel the use of steel castings, which have displaced cast iron in many places, with attendant great reduction in weight. The first use of steel castings was attended with considerable annoyance because it was unfortunately assumed, perhaps naturally, that, barring the much greater shrinkage, it could be treated very much the same as cast iron, and it was consequently asserted with confidence that anything which had been made in cast iron could be made in cast steel. This is doubtless true to-day, but it certainly was not true ten or fifteen years ago, as I know to my personal sorrow, because designs which would have been considered simple in cast iron had to be entirely changed to meet the conditions then existing for steel castings.

It may be well to mention in this connection that about the same time that steel came into use displacing wrought iron, white metal for bearings and the stronger bronzes also came into use, thus giving the designer much better materials to work with and again reducing weights. We may also mention here the gradual displacement of copper for steam piping by steel pipes, owing to the fact that for the high pressures common at present copper pipes would have to be very thick, making it difficult to secure a sound joint, and also to the serious diminution of the strength of copper by the high temperature.

FORCED DRAFT.

Forced draft dates back of course to Stevenson's *Rocket*, and its first use for marine purposes was by Mr. Robert L. Stevens on the Hudson River steamers in our own country prior to the Civil War. During that war Mr. Isherwood built a number of gunboats which used forced draft, but it had fallen into disuse until about 1882 for naval vessels, when it was introduced into the English navy and still later was applied in the merchant service.

In naval machinery forced draft has been of the greatest possible importance, because it has reduced boiler weights probably almost one-half. In the navy the natural limitations as to space and weight prevent the use of forced draft with very much economy of fuel. It is obvious that if the rate of combustion is increased from 15 pounds of coal per square foot of grate to 40 pounds, there ought to be an attendant increase of heating surface. In the merchant service, or at least in certain classes of vessels in that service, it is possible to do this, and in one of my annual reports I made a comparison between the boilers of a merchant vessel called the *Iona* and those of the *Baltimore*. In the *Iona* there were 75 feet of heating surface for 1 of grate, while in the *Baltimore* the ratio was about 30 to 1; but had the *Baltimore's* boilers been designed with any such ratio as in the *Iona*, their weight would have been almost double the weight of all the machinery of that vessel as actually built.

Mr. James Howden has made a specialty of forced draft under economical conditions, heating the air before admission to the ash-pit; and his system is now in use on most of the large steamers, the aggregate horsepower running up, I believe, to over a million.

HIGH ENGINE-SPEEDS.

About the same time as the reintroduction of forced draft in naval vessels, the improved materials and workmanship made it possible to get higher rotational speeds, and, as remarked earlier, the true conditions of propeller design being understood, there was a marked increase in the speed of rotation of the engines. Naval vessels, from the necessary limitation of keeping the vital parts of the machinery protected as far as possible, have always run faster than the engines of the merchant service, although this did not always mean that their piston speed was greater. The mistake is sometimes made of attributing lighter machinery to higher piston speeds, but unless this is accomplished by increasing the number of revolutions it will not produce that effect. In the early days, 60 or 70 revolutions per minute for what was then considered a large engine of 4,000 or 5,000 horsepower, was about the limit, but in engines of as much as 8,000 horsepower for a single set one finds the revolutions to-day as high as 130. Of course it is not practicable to show to just what extent any one line of progress has reduced weights, by comparing the machinery at periods wide apart, because the increase of steam pressure, increased rotational speeds, improved materials and better designing have all gone along together;—but it is interesting to note that in the *Warrior* of 1861, with 22 pounds boiler pressure and 54 revolutions, the horsepower per ton of machinery was 6, while in the *Minneapolis* of 1891, with 165 pounds pressure and 133 revolutions, the horsepower per ton is 10.9. From a simple mechanical standpoint, contributing agencies to the high speeds are the much more perfect journals of the steel shafts, and the superior white metals used for bearings, and the rigidity of the steel hull of the ship, as compared with the older conditions. The best makers now grind their bearings true to a mandrel which represents perfect alignment. In the old days all main shaft bearings were hollow for water circulation, which was generally needed, and there was usually provision for a spray of water on the crank pins. In the modern engines which have been well built and are carefully adjusted, there is no necessity for water even at very high speeds under full power.

THE MULTIPLE EXPANSION ENGINE.

The change from the simple to the compound engine involved a certain amount of difficulty and opposition, but the lesson was then learned pretty thoroughly that the amount of expansion in a single cylinder was moderate. Consequently as steam pressures rose the leaders of the profession became convinced that to secure adequate economy a further stage of expansion was necessary, and this brought about the triple expansion engine, the credit for which is deservedly given to Dr. A. C. Kirk, of the firm of R. Napier & Sons, of Glasgow, who first successfully used the triple-expansion engine on the steamer *Aberdeen*. The adoption of the triple-expansion engine was almost immediate, and, after the success of the *Aberdeen* was demonstrated, all new engines of any size were built on that principle. It seemed a logical extension of this idea that with still further increase of pressure there should be the quadruple-expansion engine, and a number of these have been built; but the advantage as compared with the triple-expansion engine up to the point beyond which pressures have thus far not gone, does not seem to be clearly demonstrated, and a great many designers are adhering to the triple-expansion engine with four cylinders, one high, one intermediate, and two low-pressures.

WATER-TUBE BOILERS.

Like so many other details not only in marine engineering but in other lines of work, features which are introduced in a practical way in recent times are found to have a comparatively ancient origin. This is true of the water-tube boiler, which in its recent use dates from about 1880. The excavations at Pompeii have shown small boilers almost identical in construction with some of the best of our water-tube boilers, although they were doubtless used only for a circulation of hot water.

The object of the water-tube boiler is to reduce weights, give greater safety against explosion, greater rapidity of raising steam, and an increase of economy in the generation of steam. The various makes of water-tube boilers are too numerous to mention, but they divide themselves into two broad general classes; those with straight tubes of large diameter, say four inches; and those with curved tubes of small diameter, from an inch to an inch and a half. Probably no single boiler possesses all the merits which a perfect water-tube boiler should have, and in nearly every case the attempt to secure certain advantages brings attendant disadvantages, and *vice versa*. The large straight tube boilers are not so light as the ones with small tubes; and it is more difficult to secure adequate economy, which is dependent largely upon skillful baffling. They do not permit of such rapid raising of steam from cold water as the smaller tube boilers, because, like the Scotch boiler, they carry a large reserve supply of water in the boiler after the manner of the Babcock and Wilcox boiler. On the other hand, they permit the replacement of a defective tube and of the cleaning of a tube much more readily than the tubes which are bent. Likewise it is only necessary to carry one size of spare tubes, while the bent tube boilers require several sizes and shapes.

As far as safety against explosion is concerned, there can be no doubt that there is less danger of an actual disaster affecting the whole ship,—although the worst accident which we ever had with a boiler in my naval experience was in connection with a water-tube boiler on a torpedo-boat, where all the crew of a fire room were scalded to death. Nevertheless, the boiler itself did not explode and was quite easily repaired. On the other hand, a few years before this a locomotive boiler on a torpedo-boat in Germany exploded through the collapse of the crown sheet due to low water, and not only killed all the people in the fire room, but tore up the decks and utterly ruined the boiler itself. In this connection it is a cause of sincere congratulation that since the explosion of the *Thunderer's* boilers in the English navy many years ago, there is, I believe, no record of the explosion of a large, well built, marine boiler. For naval purposes, where weight is such a great consideration, the water-tube boiler commended itself at once, and it has now become the established practice in all navies to use only water-tube boilers in new ships. Our English friends had some trouble with the Belleville boiler, but this seems to have been due to some extent to lack of familiarity with it, and other legitimate reasons. In the merchant service, where weight is not so precious, the water-tube boiler has not thus far so thoroughly commended itself to designers; and, as remarked earlier, all of the latest large vessels are still using cylindrical boilers. Some of the reasons for the hesitation to adopt water-tube boilers are that, of necessity, an installation of large power means a very large number of boilers, because the water-tube boiler does not admit of single units of great power comparable with the large double-ended cylindrical boilers. This means an immensely greater complication in the way of piping, valves, fittings, feed pumps, etc. Moreover, owing to the small amount of contained water, which is very desirable in so far as weight is concerned, the water-tube boiler is very sensitive as regards steam pressure and water level, requiring very much more care and attention than the cylindrical with its immense amount of contained water. It seems to me not impracticable that the able men who are engaged in the study of this question will finally succeed in developing a type of water-tube boiler which will commend itself for use in the merchant service as well as in the navy. Some of the boilers fitted in the United States naval ships had but six minutes of water endurance after the pumps stopped working, while one of the Babcock and Wilcox boilers adapted to our naval use has as much as twenty-five minutes endurance, which is a close approximation to the Scotch boiler.

AUXILIARIES.

In the early steamers, almost the only independent steam auxiliary was a single pump, which could be used for feeding the

boilers while under banked fires or with the engine stopped, and for pumping the bilge. The other pumps were attached to the main engine. Such things as steam capstans and winches, steam steering gear, distilling apparatus, evaporators, forced-draft blowers and electric light engines, were not dreamed of. As time went on and the size of vessels increased, steam capstans and winches and steam steering engines came in. Then it began to be found desirable, particularly for naval engines, to remove all the pumps from the main engine, leaving it nothing to do but turn the propeller, and this brought about independent air and circulating pumps and feed pumps. Further progress introduced the distiller and evaporator, the forced-draft blowers, and the electric light engine.

Most of these auxiliaries, from the nature of the case, are driven by simple engines, the pumps usually being for very slow piston speed and without expansion. The result is that the economy of these auxiliaries is naturally very low, and for some years past it has been the aim of designers to do something to either make the auxiliaries themselves more economical, or at least to utilize the heat in the exhaust steam. In some cases it has been arranged to have the auxiliaries in the engine room take their steam from the first receiver and exhaust into the second, thus, in effect, making all their cylinders part of the intermediate cylinder, as far as the steam cycle is concerned, with its attendant economy. I remember one of my former associates telling of how he had actually tried this on his ship with a saving of some six tons of coal a day, for machinery which was then working at about 8,000 horsepower. In the case of the feed pumps, arrangements have been made to turn the exhaust steam into the suction pipe, thereby having this heat carried back into the boiler, but this has not been done to any very great extent. Another plan has been to turn the exhaust from all the auxiliaries into a feed water heater through which the main feed to the boilers would go, and this has been attended by very good results.

The question has been raised repeatedly by electrical engineers that it would be a good plan to drive the auxiliaries by electric motors, on account of their very high efficiency even at fractional loads. For the engine-room auxiliaries and the boiler feed pumps it seems to me that this is unreasonable, for it means increased complication and certainly an increase in weight; and it has never been shown to my satisfaction that there would be any material increase in economy, owing to the fact that the dynamos as usually supplied on board ship are not large enough to have very economical engines, and from the fact that there are so many machines each with an efficiency less than unity in the circuit between the boiler and the final pump.

There are some auxiliaries on board ship, however, where it would seem motors could be used to advantage, notably for driving the force-draft blowers. From the necessities of the case these are usually stowed in hot and rather inaccessible places, and it is difficult to keep the engines in good adjustment. It hardly seems to me, however, that direct-current motors are well adapted to this service, as they would have to be of the enclosed or dust-proof type, which reduces their heat carrying capacity, and as they have to go in places where the heat is already very great. If alternating current machinery were installed, it seems to me that the induction motor with its extreme simplicity and ability to withstand very hard usage would be especially adapted to this work.

It is doubtless known to all of you that in modern ships the turret turning is done by electric motors. While this is an engineering matter, on board ship it comes within the purview of the ordnance officer, and therefore I have not dwelt upon it. The anchor capstan and steering engine should be electrically driven, but because of peculiar conditions existing in the navy this has not been done.

THE STEAM TURBINE.

The latest note of progress in marine engineering seems to be the advent of the steam turbine, which for some purposes has

passed the experimental stage, and has given great satisfaction. As already mentioned, a number of small vessels of the torpedo-boat class have been built with steam turbines, and this has been followed up by their use in a number of excursion steamers and cross-channel steamers between England and France and England and Ireland. The Allan Line of steamers have also decided to equip two large new steamers with turbines, and, as we all know from the technical press, the Cunard Company have had a very able committee considering the question of their adoption for the two new flyers which that company is to build,—as a result of which they have decided to use turbines in them.

The turbine has had an extended use in the last four or five years on shore for driving electrical generators, and this has been so satisfactory that the pioneer work of the Westinghouse Company in this country has been followed by the General Electric Company, the Allis-Chalmers Company, and a number of other engine builders, who seem to have reached the conclusion that for large powers at least the turbine is quite sure to supersede the reciprocating engine. For constant speed at its rated load the turbine is very economical, comparing in this respect with the most economical reciprocating engines, and its economy does not fall off so rapidly with decrease of load at constant speed as is the case with the reciprocating engine.

For marine purposes two questions have bothered those who were seeking information, one, the question of reversal, and the other, that of economy where not only the power but the speed is reduced, as is of course necessary in the propulsion of vessels. With respect to the former, different methods have been suggested and tried in different cases. In some instances Mr. Parsons has used separate reciprocating engines which are normally idle. In others, and this appears to be in his latest practice, there are reversing turbines inside the exhaust passage of the ahead turbines, so that while the ship is going ahead these turbines revolve idly, after the manner of a fly wheel, in an excellent vacuum. When the ship is to be reversed, steam is admitted direct to these turbines and secures the reversal.

With respect to economy, there is, of course, a marked falling off from that at full power, but not more so than in the case of reciprocating engines. It seems hopeless to expect that any machine will work with the same economy at one-tenth power that it does at full power, and it would be unreasonable to expect the turbine to do this when the reciprocating engine does not.

The advantages of the turbine are, the entire absence of reciprocating parts, of bearings to be adjusted, and the extreme simplicity of operation, together with the great reduction in weight due to the very high speed. Added to this is the entire freedom from danger due to priming of the boilers, the only result being a slowing down of the turbine and reduction of economy. There is also freedom from lubricating oil getting into the boilers.

Propeller design with the turbine is more difficult than with the reciprocating engine, because the conditions are entirely different from those which have hitherto obtained, and there is so little experience with propellers running at speeds of over 1,000 revolutions a minute in the case of small ships, or at 500 to 750 in the case of large ones. The Cunard propellers, it is understood, are to be limited to 180 revolutions per minute. For it must be remarked that in spite of the fact that we now have very clear and logical rules for the design of propellers under existing circumstances logically worked out, nevertheless these rules and formulæ came after the experience rather than before. This matter, however, can undoubtedly be cared for, and when more experience has been gained the design of propellers will be as easy as for existing conditions.

For naval vessels from a military standpoint the turbine has a great deal to commend it, inasmuch as the machinery will stow very well in the ship and be out of harm's way, the propellers are so small and so well immersed that there is no chance for racing even in the heaviest seas, and all questions of vibration are elimi-

nated. As already mentioned, the saving in weight is also a matter of decided value, if it can be done.

CONCLUSION.

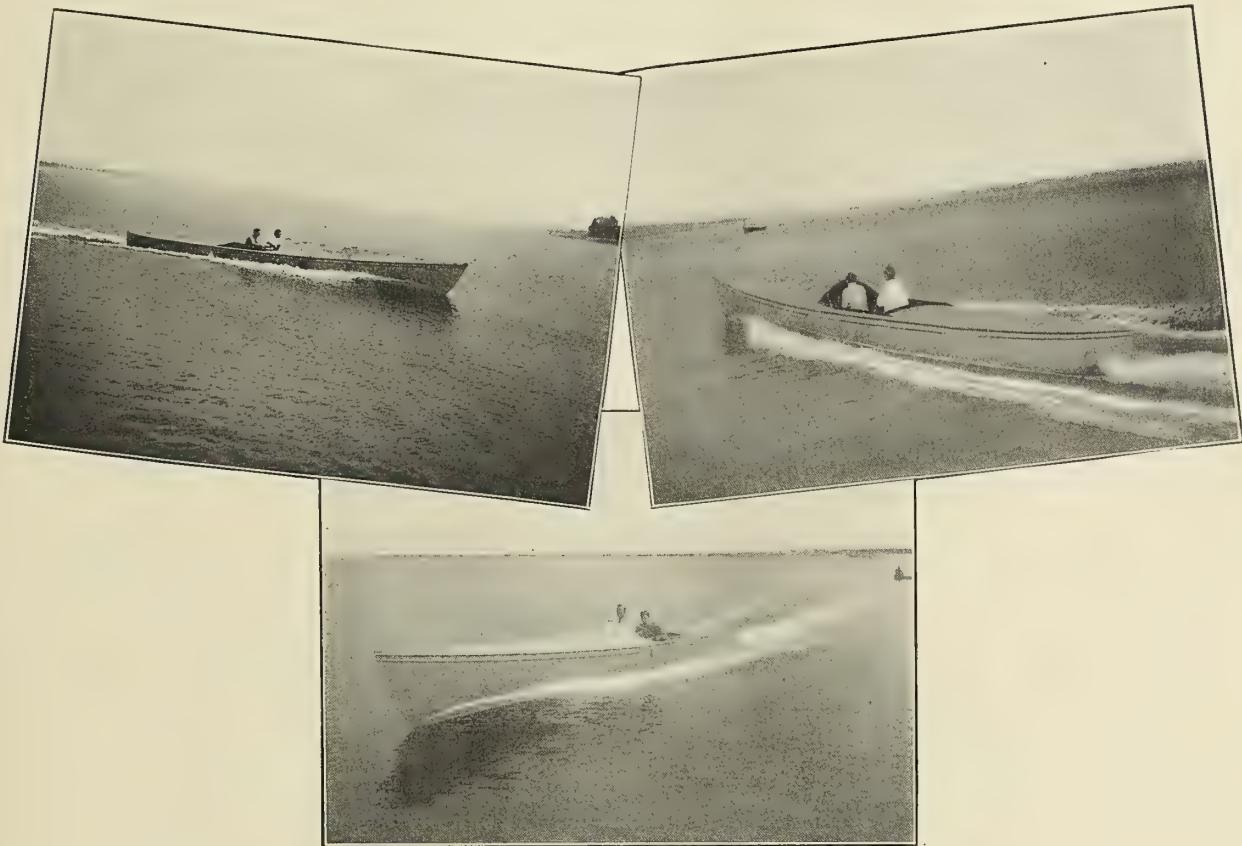
We have now considered very hastily the important epochs in the history of marine engineering, and it will be seen that they indicate steady progress along certain lines. There has been a steady increase of steam pressure, increase of rotational speed and diminution of weight, accompanied by increased economy in the making and using of steam. If time had permitted, it would have been very interesting to compare such a vessel as the *Great Eastern*, which for so long a time was the criterion of immense size, with the *Celtic* and *Cedric*, which are even larger than she was. The *Great Eastern* was simply about thirty years ahead of her time. She was a remarkable production and a great credit to

A NEW SPEED LAUNCH.

We present photograph and plans of a fast power-boat designed by Mr. Frederick S. Nock, of East Greenwich, R. I., which boat has the following dimensions:

Length over all.....	37 feet 8	inches
Length water-line	35 feet 0	inches
Beam, extreme	5 feet 0	inches
Beam at water-line.....	4 feet 4	inches
Draft to rabbet	0 feet 9 1-2	inches
Draft at propeller.....	2 feet 9	inches
Displacement	3,000	pounds

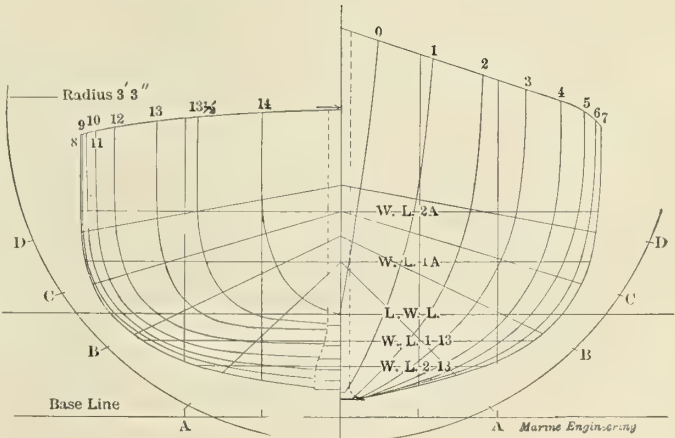
The hull, of course, is built of wood, the keel being oak, 1 1-4 by 7 inches amidships and tapered at the ends. The stem and stern posts are likewise of oak, 2 1-4 inches deep. The frames measure



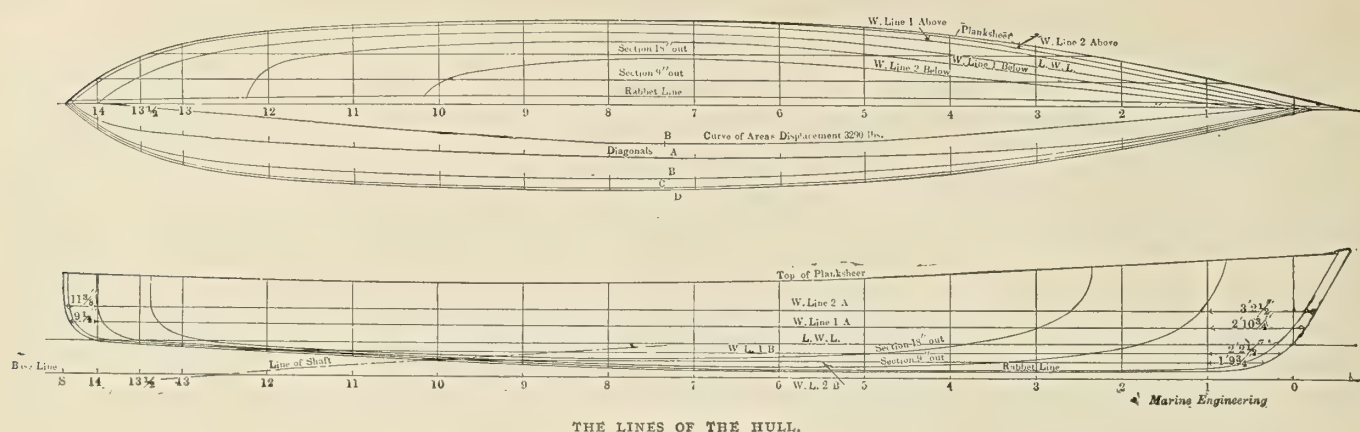
THE NEW LAUNCH AT FULL SPEED.

her designers in every way, but she was a commercial failure because marine engine manufacture had not yet reached the point where such a huge vessel could be operated profitably. She carried more than twice as much coal as one of these present day vessels, which easily makes the same speed with machinery weighing probably not more than one-third of what hers did.

The economy of the marine engine, or of the turbine if it displaces the engine, has reached a point which does not leave much room for improvement in materials and workmanship. Yet it will be unreasonable to believe that we have reached finality, for it is likely that there have been numerous periods in the past when the designers of those days could not see what the next step in advance would be, and so far as their knowledge went their design was nearly perfect. Of course there was certainly plenty of margin for increased economy with them from which we are barred, but I have no doubt that if I could live to be as old as Mr. Haswell I should see some decided improvements in the course of the next thirty years.



BODY PLAN.



THE LINES OF THE HULL.

3-4 inch square and are also of oak, being spaced 6 inches from center to center. At each alternate frame a floor is fitted. The sister keelsons are of spruce 11-2 inches wide, the upper side of the keelson being level with the under side of the cockpit floor. The fore-and-aft bearers of the engine bed are of 21-4-inch oak, being morticed over the frames and heavy floors, and securely bolted to the keel. The deck beams are spruce and the planking is white pine, 3-8 inch thick, covered with 6-ounce canvas laid on shellac. The side planking of the boat is of 1-2-inch cedar. The boat is divided into four compartments by means of galvanized steel bulkheads aft of stations 2, 8, and 13.

The shaft bearings are aluminum bronze, bolted through oak blocks fastened between frames inside of the planking. The rudder is comprised of a 3-16-inch sheet of tobin bronze with a forged tobin bronze shank, and is actuated by a quadrant.

The motor is a four-cylinder engine of the four-cycle type, manufactured by the Buffalo Gasoline Motor Company, and actuates a propeller 26 inches in diameter and 42 inches pitch. At a speed of revolution of 750 per minute the boat will make a little more than 20 miles per hour in smooth water. The gasoline supply is contained in two copper tanks situated one forward and one aft, and having each a capacity of 30 gallons. They are so arranged that the motor draws from both of them simultaneously, thus preventing any change in trim from the consumption of gasoline.

The boat has made several trips outside and has proved very seaworthy. She is not, however, a dry boat, especially in rough water with a quartering sea, but this is perhaps too much to expect in a boat of her size and speed.

It is stated that the Spring of 1905 will see the Japanese Government start the construction of two 18,000-ton battleships and two first-class cruisers of 14,000 and 15,000 tons respectively in the domestic yards.

Dealing with a Broken Crosshead Bolt.

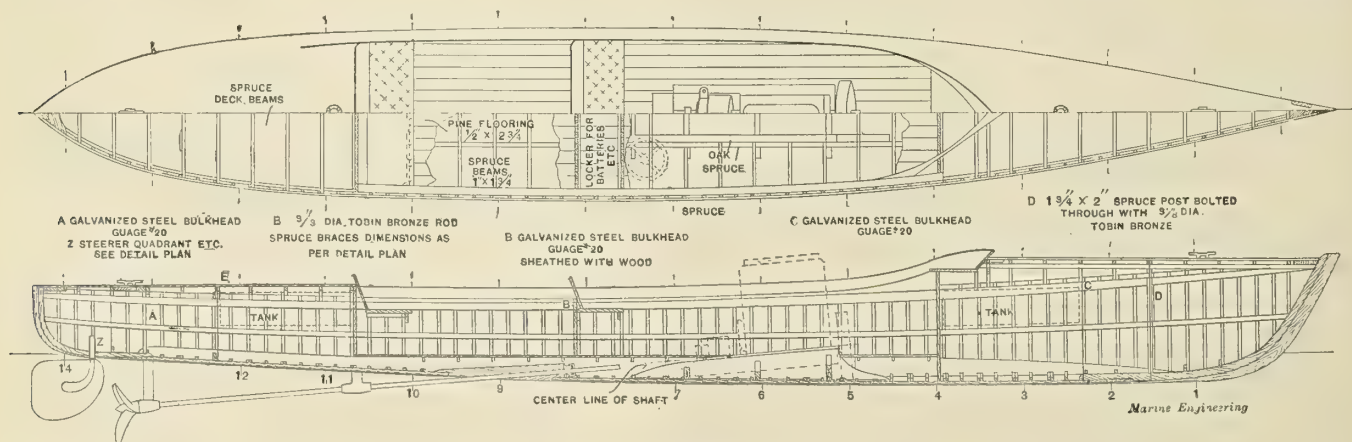
An incident that came under my notice the past summer may be interesting to some of your readers as an example of the quickness with which the marine engineer deals with difficulties which he often encounters.

The S. Y. —, on a cruise "down east" to the provinces, had run into rather bad weather, and was rolling and tumbling about as only a steam yacht can in a bad sea, when off Cape Sable a snap, followed by a jar and a succession of heavy knocks, came from the low-pressure engine. The first assistant, who was on watch, immediately shut off steam and moved the reversing gear into mid-position. By this time the chief was in the engine room.

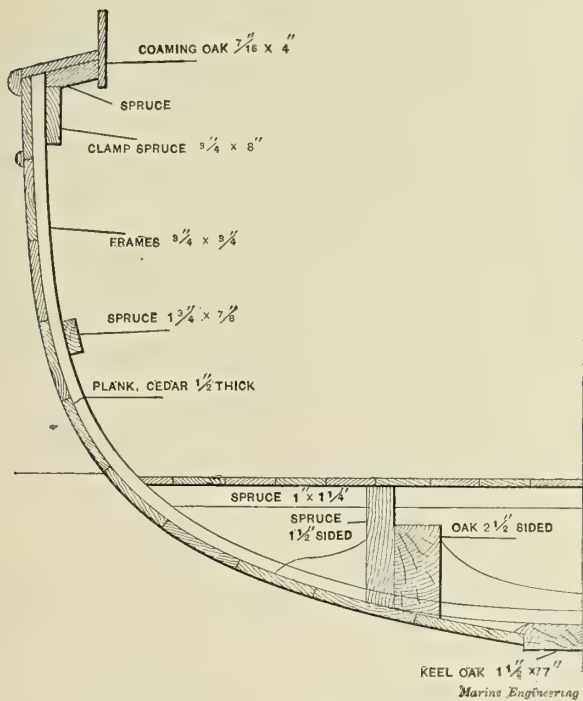
On looking the engine over, a low-pressure crosshead bolt was found snapped off and its mate badly bent, but strangely this was all the damage, everything else being intact, owing to the quickness of the engineer on watch. There were no spare bolts on board, and the yacht was in a bad place, so what was to be done must be done quickly. A part of the crowd under the first assistant removed the low-pressure connecting rod, dropped the piston to the bottom, and shoved the crosshead down from the cylinder bottom.

The chief, with another crowd, was busy at the low-pressure valve gear, and what he was doing struck me as being a bright idea; at any rate, it saved us a lot of time, and enabled us to get under way at least half an hour sooner than we otherwise would. The low-pressure had two piston valves and the chief simply disconnected them and secured them, one up and the other down, so as to make a passage through one into the top of the cylinder, and exhaust through the other into the condenser. The engines were then started with high and intermediate working, and ran in that condition until we reached Halifax. The engines were stopped only 35 minutes, and it would have been much longer had the chief stopped to remove a valve on the piston.

H. H. AUSTIN.



PLAN AND INBOARD PROFILE.



MIDSHIP SECTION OF NEW SPEED LAUNCH.

The New Cunarders.—Many interesting tests have been made in connection with the design of the new Cunarders of 25-knot speed now building. The tests were made with a large model of the vessels, to determine the form of the stern and the size of the four propellers. Previous experiments have shown that the shape given to the after part of a ship to accommodate the stern tube and shafting of the propellers materially influences the efficiency of the screws, and affects the speed for a given power. The immersed surface is increased by the bossing out or by the brackets, and the flow of water to the propellers may be interfered with. With four shafts and screw propellers the problem increases. The model was built to run in a tidal river, and had a stern easily altered to suit various suggested forms. This model was towed without propellers, and with various forms of brackets, and was also driven by propellers. The problems tested had reference to the distance of the outside propellers from the shell of the ship and their position in relation to the inner propellers, which are as close together as the circumstances permit, but the blades of the one propeller will not overlap those of the other. The arrangement arrived at will insure the greatest measure of security as there will be no overhang of shafting outside the hull.

The Coming Motor-Boat Show in Boston.

The boat show to be held in connection with the Boston Automobile Show, March 13 to 18, 1905, promises to be a record-breaker. The whole lower floor of Mechanics' Building, comprising 31,752 square feet of exhibit floor space, has been assigned for the exhibit of boats and boat accessories. Up to this writing the manager reports that 20,000 square feet has been allotted in this department. Many of the most prominent makers in the country have taken space, and applications for information and space are now in hand from eighteen other makers. Special attention is being paid to this department by the management and there is every indication to believe that it will be the most successful display of boats and accessories ever held, and as the New England states are the most fertile field for this line of business, the claim by the management that profitable results are bound to be obtained seems to be a reasonable one.

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, report under date of February 10, 1905, the following percentage of completion of vessels building for the United States Navy:

			Jan. 1.	Feb. 1.
BATTLESHIPS.				
Virginia.....	19 knots.	Newport News Co.....	75.24	76.96
Nebraska.....	19 "	Moran Brothers Co.....	64.91	65.8
Georgia.....	19 "	Bath Iron Works.....	71.94	73.61
New Jersey.....	19 "	Fore River Shipbuilding Co.....	74.3	75.7
Rhode Island.....	19 "	Fore River Shipbuilding Co.....	77.1	80.7
Connecticut.....	18 "	Navy Yard, New York, N. Y.....	61.51	64.51
Louisiana.....	18 "	Newport News Co.....	66.5	67.8
Vermont.....	18 "	Fore River Shipbuilding Co.....	31.5	35.4
Kansas.....	18 "	New York Shipbuilding Co.....	49.1	41.3
Minnesota.....	18 "	Newport News Co.....	52.84	55.7
Mississippi.....	17 "	Wm. Cramp and Sons.....	17.84	20.6
Idaho.....	17 "	Wm. Cramp and Sons.....	15.39	18.09
New Hampshire.....	18 "	New York Shipbuilding Co.....	0.0	0.0
ARMORED CRUISERS.				
Pennsylvania.....	22 knots.	William Cramp and Sons.....	98.25	98.25
West Virginia.....	22 "	Newport News Co.....	98.5	99.25
California.....	22 "	Union Iron Works.....	68.7	70.
Colorado.....	22 "	William Cramp and Sons.....	99.27	100.
Maryland.....	22 "	Newport News Co.....	95.14	96.5
South Dakota.....	22 "	Union Iron Works.....	65.9	67.9
Tennessee.....	22 "	William Cramp and Sons.....	60.53	63.13
Washington.....	22 "	New York Shipbuilding Co.....	62.7	62.7
North Carolina.....	22 "	Newport News Co.....	0.0	0.0
Montana.....	22 "	Newport News Co.....	0.0	0.0
SEMI-ARMORED CRUISERS.				
St. Louis.....	22 knots.	Neafie and Levy Co.....	58.5	58.5
Milwaukee.....	22 "	Union Iron Works.....	65.1	66.7
Charleston.....	22 "	Newport News Co.....	88.32	90.22
PROTECTED CRUISERS.				
Chattanooga.....	16 1/2 kts.	Lewis Nixon.....	99.13	99.13
Galveston.....	16 1/2 "	Wm. R. Trigg Co.....	96.	96.
GUNBOATS.				
Dubuque.....	12 knots.	Gas Engine and Power Co.....	77.6	85.7
Paducah.....	12 "	Gas Engine and Power Co.....	73.1	76.2
TRAINING SHIPS.				
Cumberland.....	Sails....	Navy Yard, Boston.....	88.	91.
Intrepid.....	"	Navy Yard, Mare Island.....	72.	78.
TRAINING BRIG.				
Boxer.....	Sails....	Navy Yard, Portsmouth.....	96.	97.
TORPEDO BOATS.				
Stringham.....	30 knots.	Harlan and Hollingsworth.....	99.	99.
Goldsborough.....	30 "	Wolf and Zwicker.....	99.	99.
Nicholson.....	26 "	Lewis Nixon.....	99.	99.
O'Brien.....	26 "	Lewis Nixon.....	99.	99.

Coming Ocean Monsters.—It has been commented on as a novelty by some of the papers that the two 35,000-ton "intermediate" 17-knot steamers that are being built for the Hamburg-American line are to have passenger hoists, as well as companionways, leading from the lower decks to the promenades. But a lift, or elevator, was fitted on one of the Atlantic liners built on the Clyde some sixteen years ago. It was ultimately discarded because the rolling of the ship caused the hoist occasionally to jam, and when the ship righted the accumulation of power resulted in a suddenly accelerated speed. There has, however, been immense improvement in lifting mechanism since then, and there may not now be trouble with a modern equipment. In King Edward's yacht there is a hoist for His Majesty's use, but it is of limited capacity. There are ammunition hoists in warships which work well. Whether a large lift, with a 40- to 50-foot travel, will work smoothly under all weather conditions remains to be seen. The *America* is being built at Harlan and Wolff's at Belfast, and the *Kaiserin Augusta Victoria* at Stettin. They will carry 3,589 passengers each, 600 first-class, 300 second, 250 third, and 2,139 in the steerage. They will have three promenades, one reserved for lounge chairs. There will be no upper berths in the first-class state rooms. There is to be an *à la carte* restaurant, separate from the dining saloon, where passengers may have what they pay for, distinct from the service included in the fare, and there will be Turkish baths. The *America* will not be ready until the autumn of 1905, and the other steamer in the summer of 1906.

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Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

Naval Appropriations.

As we go to press, word comes from Washington that the bill providing for the increase of the United States Navy, which has already been severely cut in committee, bids fair to lose one-half of its remaining strength, if not to disappear completely. The full needs of the service, as recommended by the general board of the navy, of which Admiral Dewey is the president, comprise: three battleships, five scout cruisers, six torpedo boats, six torpedo boat destroyers, two squadron colliers, and three gunboats. Realizing, however, the fact that the government was facing a heavy deficit, the general board expressed its willingness to compromise on the three battleships and six torpedo boats, allowing the scout cruisers, torpedo boat destroyers, colliers and gunboats to be eliminated from the program. But the house committee on naval affairs was not content with this reduction, and the estimate was cut down to two battleships only, and so reported to the house. Now there appears to be grave danger that even this very moderate provision will be reduced to a single battleship; while the senate seems disposed to go further, and, perhaps, prohibit all increase of the navy during the present session of Congress.

With the foreign policy of the administration demanding adequate naval backing; with the continued vigorous support of the Monroe doctrine requiring the maintenance of such a force as will give pause

to any ambitious European power hankering after South American colonization; with the naval experts who had the task of deciding upon the proper augmentation of the navy agreed upon a definite program as necessary for the case in hand—a program not at all excessive, even in its original entirety; we are provided with the not unusual spectacle of a non-technical body, calling in the advice of technical experts, and then deliberately declining to follow the advice so obtained. Unfortunately there can be no appeal from the decision, once made. The result means the loss of exactly one year in the task of providing the United States with the force necessary to preserve its position among the nations, a force all the more necessary because of the peculiar position in which we are placed by the problem of looking after the destinies of the entire western hemisphere. That this problem is self-imposed makes no difference; it has been considered requisite for the preservation of our integrity and our isolation, and has not been successfully assailed. The position is one from which we are extremely unlikely to retreat, nor is there strong reason why we should do so. We are virtually pledged to maintain the *status quo* in the two Americas (and islands appertaining thereto), and we will maintain it.

The reasons actuating the opponents of the desired increase seem to be more of a financial nature than a political. The policy of economy is advocated—a policy of extreme sanity, but, in such a case as this, of doubtful utility. And while the question of economy of expenditure is an exceedingly apt one when the disbursements of the government are greater than the income, yet this condition of affairs is but temporary; moreover, the current cash balance in the treasury, aside from the \$150,000,000 gold reserve, is nearly \$200,000,000, against which to charge the difference of (say) \$10,000,000 between the two estimates.

The situation can scarcely be considered as acute; yet it might well give rise to conditions of considerable awkwardness, if not actual humiliation. With an adequate naval force to back up our demands that the intolerable state of affairs in Cuba in 1897 should cease, we would never have had a war with Spain. It was only our apparent unpreparedness for a conflict which precipitated hostilities. That we were enormously successful against a decadent adversary is no argument in favor of a continued state of unreadiness to maintain our position in the diplomatic world. Our next opponent, if we ever have one, will not, perhaps, be so ill-equipped for the combat, and we may then awake to a realization of the folly of neglecting the opportunities which a time of peace affords—opportunities rendered all the more precious by the fact that the construction of ships now requires as many years as it did months at the time when Washington uttered the wise advice: "In time of peace, prepare for war."

On this page is depicted the work that is being done in the rehabilitation of the American merchant marine. All of the vessels building for over-sea service are fully illustrated and described.

ABSTRACTS FROM RULES AND REGULATIONS OF THE AMERICAN BUREAU OF SHIPPING.

EDITION OF 1904.

ARRANGED BY HARRISON S. TAFT.

I. Measurements.

ITEMS.		One- Two- Three- Deck Vessels.	Hurricane-Deck Vessels.	Poop; Bridge House; Topgal- lant Forecastle Deck Vessels.	Raised Quarter and Fore Deck Vessels.
Length.—From fore side of stem to aft side of rudder post; but to aft side of propeller boss in single-screw ships.	L	On L. W. L.	On L. W. L.	On L. W. L.	On L. W. L.
Depth.—From top of bar keel or flat plate keel at center line amidships.	D	To upper deck.	To deck next below the hurricane deck.	To upper deck.	To upper deck.
Girth.....	G	" " "	" " "	" " "	" " "
Breadth.....	B	Maximum beam.	Maximum beam.	Maximum beam.	Maximum beam.
First Number.—Frames, floors, and reverse frames.....	F	$\frac{1}{2} B + D + \frac{1}{2} G$	$\frac{1}{2} B + D + \frac{1}{2} G$	$\frac{1}{2} B + D + \frac{1}{2} G$	d. $\frac{1}{2} B + D + \frac{1}{2} G$
Second Number.—All other members except deck beams	S	$F \times L$	$F \times L$	$F \times L$	$F \times L$
Equipment Number.—Anchors, chains, warps.....	E	S	b. $F \times L$	c. S	S

- a. To second deck in three-deck vessels.
 - b. Depth measured to the hurricane deck; girth to the second deck, in determining the equipment number for general ocean-going hurricane-deck vessels.
 - c. To be increased by 10% of itself when combined length of said structures exceeds $\frac{1}{2}$ the vessel's length.
 - d. $\frac{1}{2} B + D + \frac{1}{2} G +$ Ht. of R. Q. D. above main deck when combined length of said structures exceeds $\frac{2}{3}$ the vessel's length.
- In vessels of 700 feet or over in length, the depth and girth are to be measured to one deck higher than for a three-deck vessel. Scantling of deck and hold beams is to be regulated by the length of their respective longest beam.

2. Scarphs.

Scarphs of bar keels, stems and stern frames should be in length not less than ten times the required thickness of a bar keel.

3. Stems.

Stems are to be of the same scantling as bar keels and they should maintain their full sectional area up to the deep load water line with a reduction to 3-4 of said area at their head. They should be rabbetted for the shell plate ends.

4. Stern Frames.

1. The lower part of stern frames forming part of the keel should extend at least 2 I-2 frame spaces forward of the propeller post in single-screw vessels but 1 I-2 frame spaces in all other kind of ships.
2. The propeller boss on each side of the hole for the stern tube should be at least 1 I-2 times the required thickness of a bar keel.
3. Rudder posts should extend to, and be securely fastened to a main-deck beam.
4. High-speed single-screw vessels are to have their propeller and rudder posts of 10 to 20 per cent. greater sectional area than otherwise.
5. Gudgeons on stern posts and rudders should be spaced not more than 5 feet apart.

5. Keels and Center Line Keelsons.

1. CENTER LINE CONTINUOUS PLATE KEEL AND KEELSON.

a. Said keel and keelson must be equal in depth to the depth of midship floor plates at the center line plus the depth of the tabulated bar keel, with side plates below the floors equal in depth to said bar keel; the combined thickness of the center and side plates being at least equal to the width of said bar keel.

b. (1). If the continuous plate keel and keelson is worked flush with top of floors, a horizontal plate stringer, in width 1 I-3 times the depth of midship floors, must be worked on top of floors at the center line; said stringer plate being clipped to floors and to top of the vertical plate keel and keelson with angle clips equal to reverse bars. If the depth of the center floors is 24 inches or over a continuous vertical plate keelson, in depth 3-4 of a specified depth, with double top angles and a rider plate, must be secured to the top of said horizontal plate stringer at center line by continuous double fore-and-aft angles, riveted back to back and embracing said vertical plate keelson.

(2). If the vertical plate keel and keelson extends above top of floors a horizontal plate stringer is to be worked on each side of said keelson. Width of each stringer plate should equal 3-4 the center depth of midship floors, and they should be secured to said keelson with continuous angles riveted back to back on top of said horizontal stringers, and embracing the top edge of the plate keel and keelson. If the depth of floors amidships is 24 inches or over, the center line keelson is to extend above top of floors at least 3-4 of a specified height; and in addition to the two above horizontal plate stringers and their continuous fore-and-aft angles, two top angles with a rider plate are to be worked at upper edge of said keelson. The vertical plate keelson may be built of two pieces, having their lap between the continuous angles on top of the horizontal plate stringers.

2. INTERCOSTAL CENTER LINE VERTICAL KEELSON.

a. Said intercostal keelson should extend above top of floors, so as to be embraced by two continuous fore-and-aft angles riveted back to back on top of floors. If the center depth of midship floors exceeds 12 but is under 18 inches, a bulb plate equal in depth to the main-deck beams is to be riveted to the top edge of the intercostal plate

keelson; being embraced by the two continuous fore-and-aft angles on top of floors. If said depth is or over 18 inches a vertical plate keelson, in depth at least 3-4 of a specified height with top angles and a rider plate, is to be worked in lieu of the above bulb plate. When said depth of floors is 24 inches or over horizontal plate stringers, each in width 1-2 the center depth of floors, are to be worked on top of floors on each side of the vertical intercostal keelson and be secured to the top of same by the continuous fore-and-aft angles on top of floors.

b. A middle line intercostal plate keelson may be worked flush with top of floors provided a horizontal plate stringer, in width 3-4 the center depth of midship floors, is worked on top of floors at the center line and is secured to the top of said keelson with double clip equal to reverse bars. Continuous fore-and-aft angle bars; said angle bars with a bulb plate; or a plate keelson with top angles and a rider plate must be worked at the center line on top of said horizontal plate stringer; as when the intercostal keelson extends above top of floors.

3. MIDDLE LINE KEELSON ON TOP OF FLOORS.

When this type of keelson is used they are to have double top angles with a rider plate, and be secured to a horizontal plate stringer on top of floors at the center line with double angle bars riveted back to back embracing said vertical keelson. Said plate stringer is to be in width 3-4 the center depth of midship floors, and is to extend for at least 3-4 length amidships. Butt straps of rider plate are to be on top; those of stringer plate on under side. When said keelson is or over 30 inches high it is to have vertical stiffening angles under all hold stanchions. Said stiffeners are to be joggled over the top and bottom bars.

4. FLAT PLATE KEELS WITH VERTICAL MIDDLE LINE KEELSONS.

(1). When the second number is or above 30,000, the flat plate keels must be doubled for 3-5 length amidships. The doubling plates are to equal the thickness of the garboards. They are to be fitted with a continuous or an intercostal vertical keelson. Flat plate keels are to maintain their midship weight for 1-2 length amidships.

(2). The middle line vertical keelson is to be double clipped to floors and be secured to the keel plates with continuous double angles for a continuous keelson, but with double intercostal clips for an intercostal keelson. The top of a continuous middle line keelson is to be worked as stated under 5-1-b; that of an intercostal middle line keelson as stated under 5-2.

5. CENTER LINE KEELSONS OF SMALL VESSELS.

They can be built of two angle bars riveted back to back on top of floors; or of a "T" bar or "T" beam section of equal area. A bulb plate may be riveted between said keelson bars in lieu of a plate keelson in those vessels whose second number is not over 5,000.

6. All center line continuous or intercostal plate keelsons, and all vertical keelsons on top of the horizontal plate stringers on top of floors, are to extend as far fore and aft as possible. The horizontal plate stringers on top of floors fitted to intercostal middle line keelsons should extend 3-5 vessel's length amidships; but when they are fitted to a "Middle line keelson on top of floors" they shall extend only for 3-4 length amidships.

7. Vertical flanges of the lower keelson bars and horizontal flanges of the top keelson bars are to be the longest.

6. Frames.

1. All frames should extend in one piece (except in double-bottom vessels) from the keel to the decks specified in Table 2. They should maintain their midship scantlings for 3-5 length amidships.

2. All water-tight bulkhead frames should be worked double. Scarph pieces when used (to be used except when a continuous middle line vertical keelson is fitted) are to be from 3 to 6 feet long, and fitted to every frame for at least 3-4 length amidships. At extreme ends of the ship the lower part of the frames should be lapped and riveted to each other.

3. When the reverse bars are cut at the margin plate of the inner bottom plating the frames are to be worked double at said point.

7. Floor Plates.

1. Floor plates are to be fitted to every frame throughout the ship. For 1-4 length amidships they must extend up the bilges above top of the keel plate to at least twice their center depth; but 2 1-2 times said depth in deep-framed vessels for 1-2 length amidships. At the lower turn of the bilge they must be at least 1-2 their center depth, and at ends equal to the moulding of the frame. Fore and aft said 1-4 length, said height at the bilge may be reduced until the floors become straight on top, but no floor is to be of less depth at the center line than midship floors. At extreme ends of the ship they are to be made deeper in order to securely fasten the sides of the ship together. Engine-room floors which are or over 1 1-3 times depth of midship floors may be finished straight on top from bilge to bilge.

2. Floors are to be secured to all center line keelsons with double clips equal to reverse bars. They are to maintain their midship weight for 1-2 length amidships.

3. Bulkhead floor plates are to extend above the ordinary floors so as to provide a sufficient lap for the riveting of the bulkhead plates to said floors above the reverse bars.

4. Limber holes shall be cut in all floors so as to provide a ready drainage for the bilge water.

5. TRANSOM FLOORS. They are to be at the center line 1 1-3 times the depth of midship floors. If less than 15 inches in depth they are to be secured to fore side of the rudder post with tap rivets, but if 15 inches or over in depth they are to have double clips, equal to the frame bars, to said post.

8. Reverse Bars.

1. Said bars are to be fitted to the top of every floor plate, and are to maintain their midship scantling throughout.

2. Reverse bars must be worked double on all floors throughout the machinery spaces and in the coal bunkers located between the boiler and engine room. Said double reverse bars are to extend from bilge to bilge.

TABLE 2.—EXTENT OF FRAMES AND REVERSE BARS.

Type of Ship.	Frames.	Reverse Bars.
One-deck vessels.....	All to upper deck.....	Under 13' depth: to upper turn of bilge and to upper deck alternately. 13' under 17' depth: to U D and to one foot above the U B S alternately. 17' depth and over with hold beams or web frames: to H S and U D, alternately.
Two- " "	" " " "	To upper and second decks alternately.
Three- " "	" " " "	c. To U D and S D alternately for $\frac{3}{8}$ length from stern. To U D on every frame for $\frac{3}{8}$ length from stem.
Ocean going hurricane-deck vessels.	" " hurricane deck.....	c. To H D and S D alternately for $\frac{3}{8}$ length from stern. To H D on every frame for $\frac{3}{8}$ length from stem.
Coastwise hurricane-deck vessels.	b. To H D and S D on alternate frames for $\frac{3}{8}$ length from stern. Forward frames all to H D.	c. To S D for $\frac{3}{8}$ length from stern. To H D and S D alternately for $\frac{3}{8}$ length from stem. Second number 25,000 or over to S D and H D alternately throughout.
In way of raised quarter or raised fore decks.	All to said decks.....	
In way of full poops and enclosed bridge houses.	a. All to said decks, ...	c. All to main deck.
In way of topgallant forecastle deck.	All to said decks.....	c. Alternately to said deck.
Sailing vessels, 20 feet or over in depth.	" " upper deck.....	All to upper deck.

H D = hurricane deck. U D = upper deck. S D = second deck (deck next below upper or hurricane deck).
U B S = upper bilge stringer. H S = hold stringer.
a. Alternate frames only to said decks when partial bulkheads are fitted in said spaces.
b. c. Extent of said $\frac{3}{8}$ and $\frac{3}{8}$ length need not exceed 60 feet from stem.
c. A double angle bar side stringer may be substituted in lieu of running the reverse bar up at said $\frac{3}{8}$ and $\frac{3}{8}$ length forward, and in poop and enclosed bridge house and topgallant forecastle deck spaces.
When a rounded gunwale is adopted the frames are to extend to lower part of said rounded gunwale.
The above hurricane-deck vessels to be of the type having a continuous hurricane deck all fore and aft.
When the height from top of deck planking to upper side of the deck beam of the hurricane deck, exceeds 8 feet, the reverse bars on alternate frames must extend to said hurricane deck in all hurricane-deck vessels. Or else a double angle bar stringer is to be worked between decks, clipped to frames, and to shell where alternate frames are not extended to hurricane deck. Or web frames may be fitted at 6 or 8 frame spaces apart, secured to deck beams at top; and to deck plating at bottom with clips equal to reverse bars.

TABLE 3.—EXTENT OF REVERSE BARS IN DEEP-FRAMED VESSELS.

One- and Two-Deck Vessels.	Three-Deck Vessels.	Hurricane-Deck Vessels.	In Way of a Raised Quarter or of a Raised Fore Deck.
All to upper deck.....	To upper deck and to top of the second deck stringer angle bar alternately.	To top of the second deck stringer angle bar on all frames.	All to said decks.

9. Bilge Floor Keelsons and Side Stringers.

1. BILGE KEELSONS.

All vessels are to have bilge keelsons of two angle bars, riveted back to back, at the upper and lower turns of the bilge.

2. FLOOR KEELSONS.

a. All vessels whose beam exceeds 32 feet are to have a double angle bar floor keelson midway between the center line and the lower bilge keelson with intercostal plates riveted between the two keelson angles. Said intercostal plates are to have single clips to floors and to shell of reverse bar size.

b. Vessels of 20 and less than 32-foot beam without said intercostal plates are to have intercostal wash-plates in lieu of same, secured to the shell plating with a single clip equal to reverse bars.

3. SIDE STRINGERS. See Table 5.

4. When the bilge or the floor keelsons with intercostal plates are to be reinforced with a bulb plate, in lieu of said bulb plate one of the keelson angles may be changed to a bulb angle if the intercostal plates are made two pounds heavier than otherwise.

5. Discontinuous side and bilge angle keelsons and plate side stringers should lap each other sufficiently so as to maintain the longitudinal strength of the ship.

10. Floor Clips.

Floor clips for all plate stringers and keelson bars should be fitted on every frame for at least 3-4 length amidships in addition to the regular reverse bars.

11. Continuity of Stringers.

All the middle line, floor, and bilge keelsons should be worked continuous through the water-tight bulkheads, with water-tight collars on each side of said bulkheads.

12. Web or Plate Side Stringers.

Said stringers must be clipped to the shell with single clips and have a continuous angle bar at the face of the frames riveted to the stringer and to the reverse bars or clips on the frames. At ends of the ship flanged plates may be used in lieu of said shell clips. Said side stringers are to be worked continuously through the water-tight bulkheads and they shall be secured to same with double clips equal to reverse bars. Bracket knees of the same weight as the stringer plate are to be fitted at alternate frames, being clipped to the stringer plate with single clips equal to reverse bars. A single face bar is to be worked on the inner edge of the stringer plate when said plate is 30 inches or less in width. Otherwise a double face bar is to be used. The deep flanges are to be worked vertically for single face bars and horizontally for double face bars. Gusset plates are to be worked at each hold beam.

13. Web Frames with Deep Side Stringers in Lieu of Hold Beams.

The web frame plating should be at least of the same thickness as the standing flange of the frame bar to which they are attached, and be in width not less than 3 1-2 times the depth of said flange. They should have double reverse bars on the inner edge; the doubling bar extending on top of floors to one foot beyond the bilge keelson. They should be cut at all plate stringers, and have double clips to same, equal to reverse bars. They should be spaced at 3-4 of the spacing of the hold beams they are substituted for; but in no case need they be spaced closer than 8 feet.

14. Deep Framing for Vessels in Lieu of Web Frames.

1. The deep frame reverse angles are to have at least a 2 1-2-inch lap to the frame, but not less than 3 inches if the first number is 67 or over.
2. The plate side stringers are to be clipped to the shell with single clips equal to reverse bars. Continuous double angle bars are to be worked at the inside of the frames, being riveted to the reverse frame and to a reverse clip on each frame, and on the inner edge of the plate stringer. When the second number is 42,000 or above, a face plate, all fore and aft, is to be worked on the face bars on the inner edge of the plate stringer.
3. When the side stringers are cut at a water-tight bulkhead, they shall have bracket plates 3 pounds heavier than the stringer plates to same. Said bracket plates are to be in a fore-and-aft direction, not less than 1 1-2 times the width of the stringer plate; and on the bulkhead inside of the stringer plate not less than twice the width of said plate; with double clips to the bulkhead equal to reverse bars.

TABLE 4.—NUMBER OF PLATE SIDE STRINGERS.

Depth from top of floors to top of lowest beam tier amidships....	Under 15 ft.	15 ft. under 20 ft.	20 ft. under 25 ft.
Number of plate side stringers....	1	2	3

15. Deck and Hold Beams.

1. Said beams should be located vertically over each other on alternate frames, except as stated below.
2. Beams for a wood deck, or for a steel deck covered with wood, are to be placed on alternate frames.
3. Beams for a steel deck of 14 pounds or less, with no wood deck laid thereon, are to be spaced on every frame; but on alternate frames when said plating is 14 pounds or above.
4. Beams over 70 feet in length must be spaced on every frame.
5. Beams should be worked on every frame if possible in the engine and boiler spaces.
6. Beams without decking, except hold beams; are to be 1-8 deeper than otherwise.
7. Beams under the windlass, capstan, at mast partners, etc., are to be of midship size.
8. Beams at the ends of hatches or other deck openings over 12 feet in length should be 1 inch deeper than otherwise; the fore-and-afters being of the same depth as the end beams.
9. Web or plate beams are to have double angle bars worked on the lower and upper edges.

TABLE 5.—ARRANGEMENT OF DECK BEAMS AND HOLD STRINGERS.

Deck Beams.		Hold Beams and Stringers.		
a. Depth to top of Upper-Deck Beams Amidships.	b. Number of Tiers of Deck Beams on Alternate Frames.	Depth to Lowest Tier of Regular Deck Beams.	Beam Spacing.	Type of Stringer.
Under 24'	1	12' under 14'		D A
24' " 32'	2	14' " 16'		D A + B P $\frac{3}{8}$ L
32' " 40'	3	16' " 18'	6 Fr., 12 E	Plate
40' " 48'	4	18' " 21'	4 Fr., 10 E	"
.....	"	21' " 24'	Alt., 8 E	"

D A = double angle; B P $\frac{3}{8}$ L = bulb plate for $\frac{3}{8}$ length amidships; Alt. = on alternate frames; 8 E = on every eighth frame if extra heavy beams are used; plus sign (+) means "with a."

a. To second deck in hurricane-deck vessels. b. Not including hurricane-deck beams.

Depth to be measured from top of floors or inner bottom plating.

When web frames are used hold beams may be fitted further apart than herein stated.

When depth to top of midship hold beams is or over 12 feet a hold side stringer must be fitted midway between upper bilge stringer and hold beams.

16. Beam Knees.

Beam knees should be in depth not less than 2 1-2 times the depth of the beam to which they are secured, and not less than 1 1-2 times the depth of said beam across their throat. Plate knees should be 2 pounds thicker than their respective beams. In deep-frame ships, knees of the lowest deck beams are to be three times the depth of said beams.

TABLE 6.—ARRANGEMENT OF STEEL DECKS.

Second Number.	Under 19,000	19,000 under 25,000	25,000 under 30,000	30,000 under 42,000	42,000 under 48,000	48,000 and above
Number of steel decks.	No steel deck required.	One partial.....	One complete.....	One complete. One partial.	Two complete.....	a Three complete.

a Lower deck to be of at least 14 lbs. plating.

The Engineering Personnel in the Navy.

Editor MARINE ENGINEERING:

As a preface to his remarks on the engineering personnel of the navy, Admiral Rae in his last annual report makes the following indisputable and significant assertions:

"1. A modern navy without an efficient engineering personnel is deficient in one of its prime requisites, and will fail in the hour of battle.

"2. The present condition of engineering in the navy is a cause of grave anxiety to all conversant with the subject.

"3. No man can become an efficient engineer without a proper training and experience.

"4. The requisite training and experience cannot be obtained by serving for a few months only, and at non-consecutive periods of time, in the engineer departments of ships in commission."

To a person conversant with engineering matters as they now obtain in the navy, these statements are fraught with much that is of vital importance. Briefly stated, the machinery of the ships of the navy now in commission is kept going in a trust-to-luck manner by a handful of experienced engineer officers, a varying but small number of inexperienced and unenthusiastic line officers, and an overworked and disgruntled corps of warrant machinists. Frequent recourse to the services of the repair shops at navy yards, and the fact that the majority of the vessels are comparatively new, are the principal deterring agencies which postpone the inevitable crisis in engineering matters. The policy of the administration appears to be to keep the entire fleet in a condition of preparedness for any emergency which might arise, and aside from the conditions existing in the engineering department our vessels are undoubtedly in a high state of efficiency. But with each commander's commission handed to a former officer of the old engineer corps this state of affairs is growing worse instead of better. What will happen when all these officers are retired from sea duty is a matter of the gravest conjecture; that time is not far distant, and something must be done. The hallucination that every line officer could be made a capable engineer officer by merely spending a few months in an engine room and telling the machinist of the watch to "Make it so," has disappeared even from the minds of the fathers of that legislative monstrosity, the so-called "Personnel Bill." The remedial measures proposed by Admiral Rae for the enlargement and training of the commissioned personnel would no doubt be of great benefit, but would they, even in their entirety, furnish a complete solution of the problem? Assuming for the moment that these measures would result in an efficient engineer corps, what has been done in regard to their being promulgated? We hear of a proposed re-organization of the personnel of the navy (a bill recently introduced in Congress) which will provide for various vice-admirals, early retirements from sea duty of captains, commanders, etc., but where is the legislation for the most vital question which confronts the navy to-day? *Tempus fugit*, and the crisis is close at hand. If nothing is to be done for the commissioned personnel from an engineering standpoint, and present appearances would give one that impression, why not make the best of the present situation and do something for those who are now acting as engineers, the present corps of warrant machinists? The officers of the departed "Engineer Corps" forsook the title of "Engineer" for the higher sounding appellations (from a social standpoint only) of "Lieutenant," "Commander," "Captain," "Admiral," etc.; why not give it to those who are now doing the engineering work? "Warrant machinist" means nothing. Call them assistant engineers and be done with it. Can anyone deny the relative importance of the man in charge of an engine-room watch, especially when compared with a boatswain, gunner, carpenter, and even those relics of a musty past, the sailmakers? Then by all means give the "man below" the privilege of aspiring to that reluctantly granted title of "Withbutafter" ensign; but please note that the emphasis is always on the "after." Then, too, why not increase the number of these hard-working men, so that eventually all will not become

web-footed by constant service at sea? Going to sea is not a bed of roses by any means, and warrant machinists, as well as commissioned officers, have wives and sweethearts whom they would like to see occasionally. Because nothing can seemingly be done to reorganize a competent commissioned engineer corps, there is no good reason why every effort should not be made toward making the best of existing circumstances, and the first step would seem to be to make the present warrant machinists satisfied with their lot by granting their reasonable requests. No man dissatisfied with his position can or will fill the same efficiently. All true friends of the navy will, I feel sure, agree with me that something will have to be done for the engineering side of the navy, and that soon. While I may be classed as a pessimist for thus expressing my views, all must admit that the situation is a grave one. The voice of the press is the one that is listened to most attentively in this great country of ours, and I appeal to you to sound the alarm through the columns of your paper, which, without any intention of flattery, I believe to be the leading exponent of engineering matters, from a marine standpoint, in this country.

Very truly yours,

EX-VOLUNTEER, U. S. N.

San Francisco, Cal., Feb. 2, 1905.

OBITUARY.

We have to record the death, on the 25th of January, of Mr. Edward H. Mullin, of Millburn, N. J. Mr. Mullin was very prominently connected with electrical engineering work, both by his position with the General Company in New York, and his membership in the American Institute of Electrical Engineers. An Englishman by birth, he was a graduate of Queens College at Belfast and of the University of Edinburgh.

ENGINEERING SPECIALTIES.

Electro-Lifting Magnets.

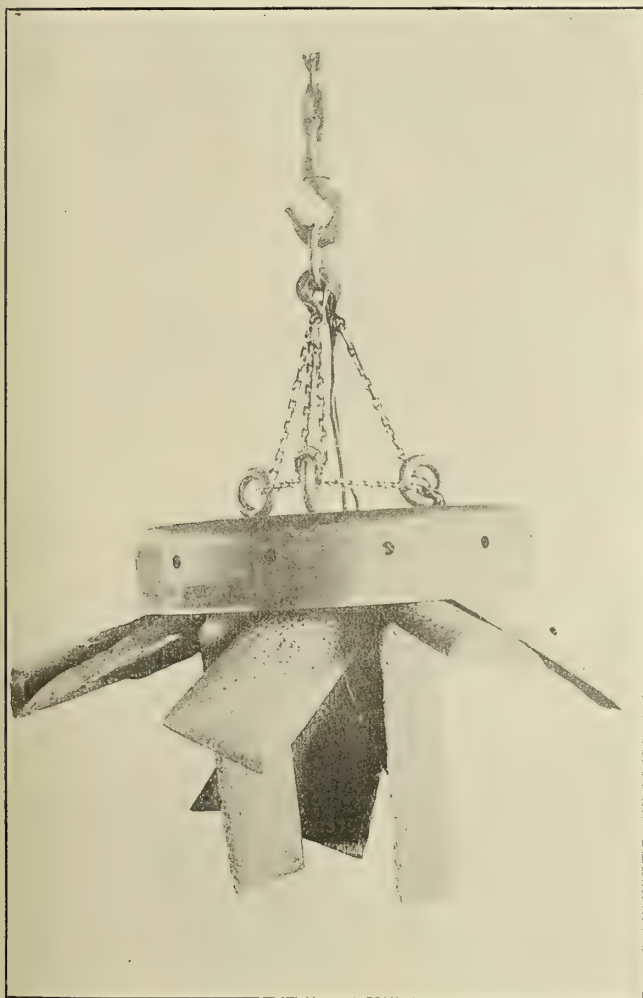
Wherever steel or iron is handled in quantities by means of cranes, a magnet will effect economies in time and labor sufficient to pay for itself in from one to six months. This statement is borne out by experience in steel mills, jobbing houses, safe works,



etc., and is evident when the methods, chain versus magnet, are compared. In all cases an electro-magnet is used, suspended from the hook of the crane, direct current at any of the common voltages being employed to energize the magnet. A flexible twin conductor cable is used to convey the current to the magnet and a

small switch operated by the crane man is usually the only additional apparatus necessary. The amount of current used is small, being from one to twelve amperes according to the service for which the magnet is designed.

In operation the magnet is lowered upon the material to be lifted, and the switch closed, thus causing the magnet to attract and hold the material, which may then be hoisted by the crane and transported to the desired point. By simply opening the switch the material is instantly released. Comparing this method of operation with the common methods of connecting the load to the hook of the crane, with chains, hooks or clamps, the saving in both labor and time is apparent, as, in general, the attachment of the magnet to the load, as well as the release of the load, may be accomplished by the crane operator without assistance, thus saving the labor of one or more men for prying up the material, attaching hooks and chains at the point of loading, and additional men at the point of delivery for unhooking the load from the crane.



Magnets can be so quickly attached to and detached from a load that by their use the work which may be done by a given crane is greatly increased—in some cases more than doubled. It frequently occurs that the attachment of lifting magnets to existing cranes so increases their capacity for handling material that the purchase of additional cranes for handling an increased output is rendered unnecessary.

Again, lifting magnets require much less head-room than hooks or chains for lifting material of considerable width such as steel plates. Therefore, by the use of magnets material can be conveniently piled to a greater height in the storage space under a given crane than is possible when chains are used, thus increasing the capacity of a given storage space without altering the crane runway or increasing the size of the building.

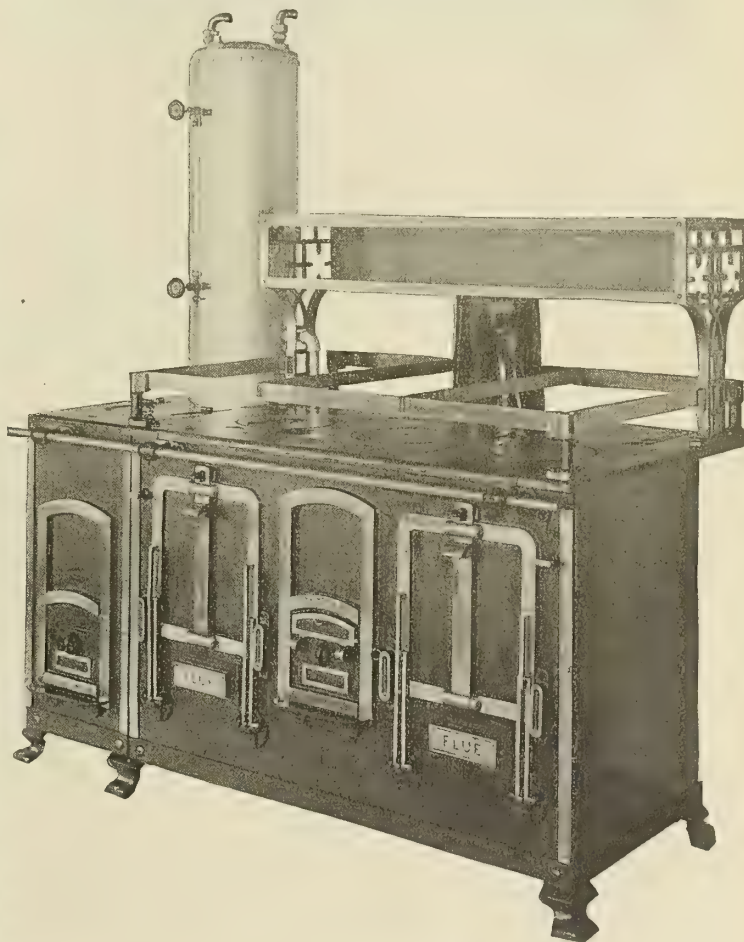
Lifting magnets may be used to great advantage in handling pig iron, scrap, rivets, bolts and similar articles in bulk, as shown in the illustrations. Ingots, blooms, slabs, billets, bars, plates, rails, structural shapes, pipe, etc., may also be handled to advantage. The Illinois Steel Company use fourteen magnets in their South Chicago works alone, and have found the economy in time and labor so substantial that they have decided to use them wherever possible in all their works.

A single design of magnet is not adapted to handling the full range of material above mentioned; on the contrary, the magnet must in every case be designed to meet the form of material to be handled. For instance, there is a wide difference in the design of a magnet for lifting ingots or blooms and one adapted to the handling of thin plates. A magnet which would handle five tons in the form of an ingot might not handle 500 pounds in the form of thin plates. It is therefore necessary to understand in each case the operating conditions with special reference to the form and range of material to be handled.

The question of risk of accident rises. The Electric Controller and Supply Company, Cleveland, O., report that in their experience with scores of lifting magnets in successful operation they have yet to learn of a single accident which has occurred through their use; while, on the other hand, accidents due to the slipping and breaking of hooks and chains are known to be of very frequent occurrence. Magnets are always built and tested to from four to five times the specified load.

Webb Galley Range.

We reproduce herewith a cut of a type of galley range manufactured by Elisha Webb & Son, Philadelphia, Pa., which is rapidly gaining favor with the ship owners because of the con-

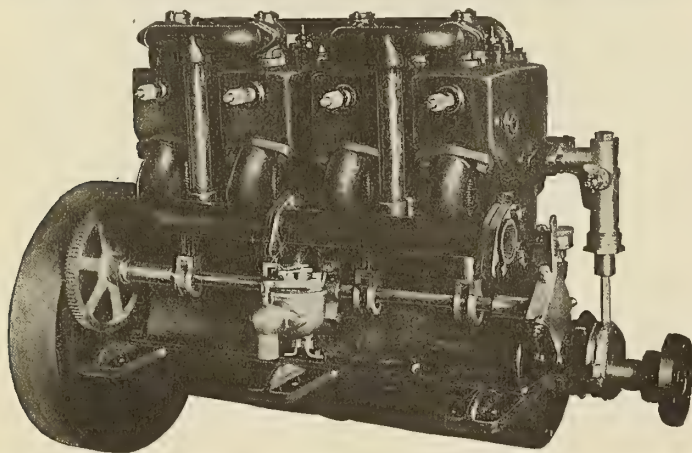


venience its use insures where installations are made. The apparatus consists of a modern steel range, with coal box, having a circulating hot-water boiler used in connection with the water-

back in the fire-box, which by an ingenious system of piping insures hot water in any desired quantity at any place on shipboard, under a pressure or gravity system. These ranges have been installed by the makers on a number of vessels of the Standard Oil Company, the 20-knot passenger steamers *Penn* and *Lord Baltimore*, harbor-master's tug *M. S. Quay*, and dredges *Atlantic*, *Benyaurd* and *Burton*. The cut reproduced shows the type of apparatus made for and installed on the five United States light vessels built by the New York Shipbuilding Company, which consisted of a 4-foot, two-oven range and circulating hot-water boiler of fifty gallons' capacity, making a complete and very satisfactory galley outfit for a vessel of this class.

The Hoffman Motor.

The design of the Hoffman motor, built by the H. L. Hoffman Motor Company, of 394 Wabash avenue, Chicago, Ill., is the outcome of years of experience. Each cylinder head, water jacket, and valve chamber is cast all in one piece. The crank chambers,



cast in halves, are bolted on, and provided with doors for inspection. The flywheels are outside. Under these specifications a motor can be built very inexpensively, and will stand hard service. The motors are all operated on the four-cycle system.

The four-cylinder motor shown in illustration develops 12 horse power with cylinder 4 inches in diameter and a 5-inch stroke. The weight of the engine complete is stated to be 550 pounds, and it is designed for the use of launches from 25 to 35 feet in length.

The single-cylinder engine has also a 4 by 5 cylinder, developing 3 horsepower. The weight is 210 pounds, and the designed service the propulsion of launches from 16 to 21 feet long.

TECHNICAL PUBLICATIONS.

The Centrifugal Pump, Turbines, and Water Motors: including the Theory and Practice of Hydraulics. (Specially adapted for engineers). By Charles H. Innes, M.A., Lecturer on Engineering at Rutherford College, Newcastle-on-Tyne. Fourth and enlarged edition. Pages 340. Figures 276. Price \$2.00. 1904. Manchester: The Technical Publishing Company, Limited, and New York: D. Van Nostrand Company, and The Derry-Collard Company.

Commencing with the assumption of the law of the conservation of energy, and afterwards assuming the law that the change of angular momentum is equal to the angular impulse of the force producing it, and that the losses of energy in a stream are proportional to the square of the velocity of flow, the author has developed the theory of operation and methods of design of the machines included in the title. The first few chapters are devoted to the exposition of the fundamental principles of the science of hydraulics. In the sixth chapter, however, the author, having "always found that theoretical principles, although absolutely necessary, are best administered in small doses, turns from them to describe a few types of hydraulic engines for producing rotation." Carried along with each description are the theory governing the case in question, and its application to the design in hand. The turbine is taken up both graphically and by

methods of calculation, the latter being given the preference on account of its greater accuracy, and only slightly increased demand upon the time of the engineer. Under the head of "impulse turbines" is given the Pelton wheel, so common in the United States, and so rarely met in Europe.

The theory of the centrifugal pump is based on the assumption, for the sake of simplicity, that the axis of rotation is vertical, it being obvious that, otherwise, "particles at equal distances from the shaft would have different velocities and be under different pressures, which would complicate the theory, although the effect in practice is unimportant." Comparisons are made, for both centrifugal pumps and turbines, between theory and the results of experimental practice, a large number of experimental results being tabulated and represented by the usual curves of performance.

Chapters are given to the more or less closely allied subjects of the steam turbine, and the centrifugal fan, while one of the last chapters in the book is devoted to a description of the plant and hydraulic apparatus of the Niagara Falls Power Company. The only steam turbine accorded extended mention is the Parsons.

The numerous cuts, which serve to illustrate and explain the text, are quite clear, though many of them are relatively crude. The typography is good.

Record of American and Foreign Shipping. Pp. 1,294. Size 8 by 9 1-2 inches. Price \$15.00. American Bureau of Shipping, 66 Beaver street, New York.

The price of this book was given in our February number as \$5.00, which is incorrect.

Types and Details of Bridge Construction. Part I.—Arch Spans. By Frank W. Skinner, M. Am. Soc. C. E. Pages 299, and 250 illustrations. 1904, New York: McGraw Publishing Company.

The purpose of this book is to present the development of advanced practice and standard details, to illustrate the classes of structures adapted to different conditions, show some of the characteristic differences between American and foreign design, besides recording important and well-known examples so as to have their principal data readily accessible. The first consideration in the selection and preparation of this data has been in every case to show clearly the special and important features, to give only what is essential to the design, and to present the requirements, conditions, and methods involved in the construction and erection of the work. The list of structures described is by no means complete, and the classification is not absolute, but both are sufficient to illustrate different types and to show a wide range of practice and detail. The bridges have been arranged in order and grouped in classes, and the descriptions are in many cases supplemented by specific references to more extended articles in technical journals or professional papers. Typography and illustrations leave little to be desired.

Maver's Wireless Telegraphy. By William Maver, Jr. 216 pages; 123 illustrations. 1904, New York: Maver Publishing Company. Price \$2.00.

The subject has been treated from both the theoretical and the practical standpoint in language as free as possible from mathematical formulæ, and the whole written in a manner designed to be clear to the average reader. The intention has been, in order to economize in space, to limit the descriptions of systems and apparatus to those in actual operation. The few exceptions to this rule relate to peculiar types of systems of which a brief account was thought to be useful. The aim has been to give a comprehensive statement of all that appertains to the art at the present time, and in the hope of supplying a complete and practical handbook of wireless telegraphy. The first part of the book was written when there was but one wireless system in operation. It may, therefore, be said that the book has grown up with the art of which it treats. This has necessitated considerable cross reference in the text, and has resulted in the inclusion in the volume of an unusually complete index.

QUERIES AND ANSWERS.

All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the editor in this column. All communications must bear the name and address of the writer.

Q. 272.—(1) If a cylindrical shell be subject to an internal pressure of 100 pounds to the square inch, and being 40 inches in diameter and 120 inches long, what will be the stress per square inch on the longitudinal and circumferential seams; also, which one of these two has to make the stronger?

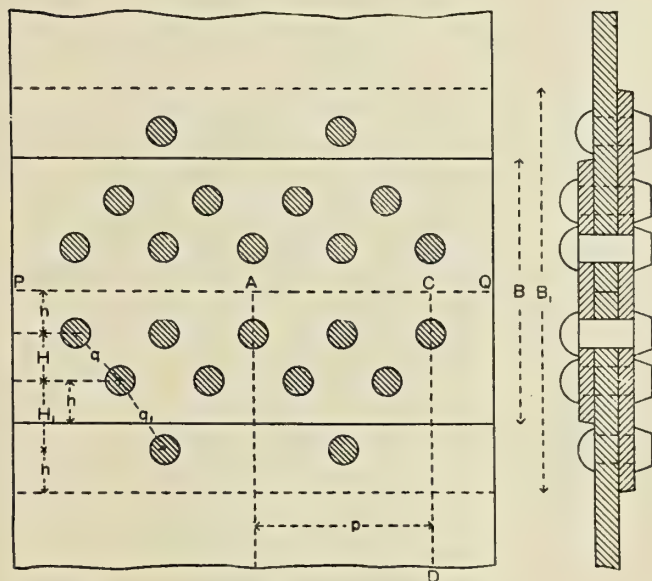
(2) Taking a rivet in single shear at 40,000 pounds and in double shear 75,000 pounds per square inch of section, please give method of analysis of butt seams, double strapped, having two rows of rivets in double shear, and one row in double pitch and single shear as below. Plate, 65,000 pounds tensile strength per square inch; thickness, 1-2 inch; thickness of strap, 5-16 inch; diameter of rivet holes, 13-16 inch; single pitch, 3 1-4 inches; double pitch, 6 1-2 inches. OILER.

A.—(1) The stress along the longitudinal seam of a cylinder subject to internal pressure is represented by the pressure per square inch multiplied by the diameter of the cylinder and divided by twice the thickness of the shell. For a circumferential seam, the stress per square inch is represented by the internal pressure in pounds per square inch multiplied by the diameter and divided by four times the thickness of the plate. It thus appears that it is necessary to make longitudinal seams stronger than circumferential seams, because of the fact that under similar conditions the stress will be greater. In the example given, the thickness of the plate is not mentioned. Assuming it to be 1-2 inch, it follows that the stress per square inch in the longitudinal seam equals:

$$\frac{100 \times 40}{2 \times 1-2} = 4,000 \text{ pounds per square inch,}$$

while the stress on the circumferential seam will be:

$$\frac{100 \times 40}{4 \times 1-2} = 2,000 \text{ pounds per square inch.}$$



(2) The strength of a riveted joint is determined by the weaker of the two elements making up that joint, the plate, and the rivets. These are usually made about equal in strength; where any difference is made between them the plate is made the stronger because of the fact that it is more subject to corrosion. A riveted joint may fail in any one of four ways: (a) by tearing out the plate along the line of rivets; (b) by shearing off the rivets; (c) by splitting the plate between the rivet holes and the edge; and (d) by splitting the plate in diagonal lines along a row of "staggered" rivets. In the present case only the first two are applicable.

(a) Let us take for consideration a section of length, p , which is just like every other such section along the seam. In that section we find that there are two places in each of which two rivet holes are in line. This reduces the effective area of the plate to:

$$t \times (p-2d),$$

where t is the thickness of the plate in inches, p the pitch of the rivets or distance between their centers, and d the diameter of the rivet holes. If we multiply this expression by the tensile strength of the material, T , we have as the strength of the plate in its weakest section:

$$T \times t \times (p-2d).$$

With the data given, this figures out as:

$$65,000 \times 1-2 \times (6 1-2 - 2 \times 13-16) = 158,438 \text{ pounds.}$$

(b) In the section in question there are four rivets in double shear and one in single shear. As the rivet holes are 13-16 of an inch in diameter, the rivets would be made 3-4 of an inch, allowing 1-16 of an inch clearance for the insertion of the rivets when hot. The cross sectional area of each rivet is:

$$3-4 \times 3-4 \times \frac{\pi}{4} = 0.442 \text{ square inch.}$$

Thus the five rivets in question will have a shearing strength represented by:

$$0.442 \times (40,000 + 4 \times 75,000) = 150,280 \text{ pounds.}$$

It is thus seen that general practice has been followed in making the plate a little stronger than the rivets.

Q. 273.—Will you please figure the horsepower on the following gasoline engines, and explain it in MARINE ENGINEERING?

The engines are of the 4-cycle type and of the best makes. (1) 2 cylinder; diameter, 7 inches; stroke, 9 inches; revolutions, 320. (2) 2 cylinder; diameter, 8 1-8 inches; stroke, 10 inches; revolutions, 350. (3) 4 cylinder; diameter, 6 inches; stroke, 8 inches; revolutions, 300.

We do not know the M. E. P. on the cylinders, but figure them at 80 pounds per square inch. GASOLINE ENGINE.

A.—(1) The area of this piston figures out at 38.5 square inches, which with 80 pounds mean effective pressure would give 3,080 pounds as the mean pressure on the piston during the power stroke. This being a four-cycle engine, and having two cylinders, there would be one power stroke for two revolutions. As there are 320 revolutions per minute, and the stroke is 9 inches, we find that the number of foot-pounds per revolution figures out at 1,155, and the foot-pounds per minute 369,800. This gives 11.2 horsepower.

(2) The piston in this case has an area of 52.85 square inches and a total average pressure during the power stroke of 4,228 pounds. As before, there is one power stroke in every two revolutions. The revolutions numbering 350 per minute with a 10-inch stroke, the work in foot-pounds per revolution figures out at 1,762, which gives 616,000 foot-pounds per minute, or 18.65 horsepower.

(3) In this case the piston area is 28.25 square inches, the pressure on piston 2,260 pounds, and as there are 4 cylinders there will be one power stroke in every revolution. The revolutions per minute number 300 while the stroke is 8 inches. The work per revolution is, therefore, 1,507 foot-pounds. This gives 452,100 foot-pounds per minute, or 13.7 horsepower.

Q. 279.—Will you kindly give me the required information through your Q. & A. column as per the following questions?

Give the drawing-board delineation of the method used in laying out worms and worm wheels, used for turning gear, together with the rules regulating the proportions of teeth and thread, and the depth of same, etc.

What is a good hand book on machine design?

G. F. M.

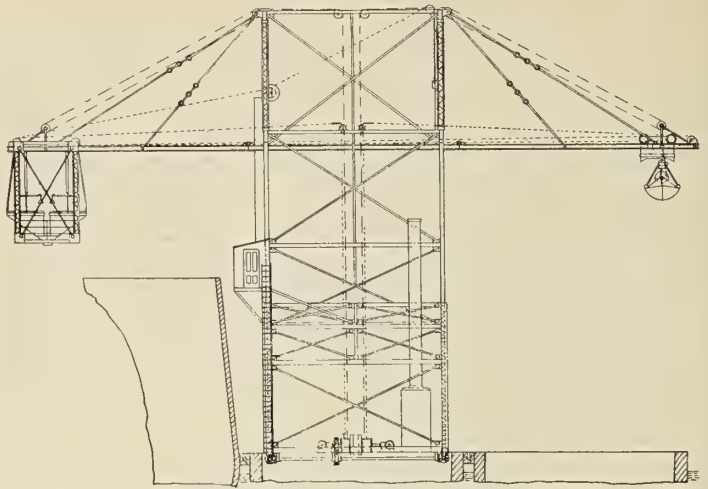
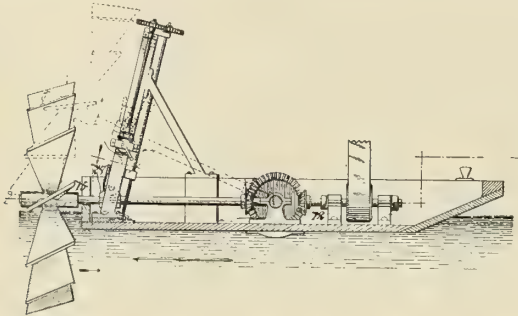
A.—The subject of worms and worm-gears is taken up on page 395 of "Machinist's and Draftsman's Handbook," by Peder Lobben, published by Van Nostrand Company, New York, in 1900, the price of the book being \$2.50. This appears to be a good book on the general subject of machine design from the standpoint of the draftsman.

SELECTED MARINE PATENTS.

775,164. APPARATUS FOR LOADING VESSELS WITH COAL OR OTHER MATERIAL. CHARLES BROWN, LIVERPOOL, ENGLAND, ASSIGNOR OF ONE-THIRD TO WILLIAM HAROLD WATSON, GREAT CROSBY, ENGLAND.

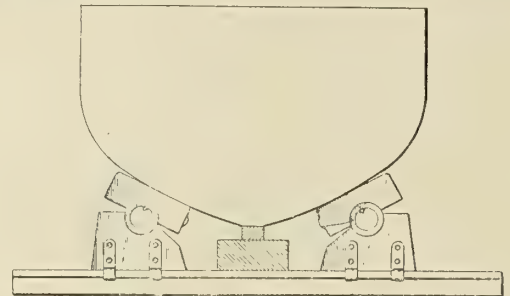
Claim.—1. In an apparatus for loading vessels, the combination of a barge for holding the material to be loaded; means on the barge for holding it at a given distance from and in rolling contact with the vessel which is being loaded; an elevator adapted to raise the material; chutes adapted to guide said material from the elevator; a vertically-adjustable hopper designed to receive the material from the chutes; and means for feeding the elevator buckets up to the material in the barge.

7. In an apparatus for loading vessels, the combination of a barge for the material to be loaded; an elevator capable of travel longitudinally and laterally on said barge; means for moving said elevator; a third adjustable wheel near the lower extremity of the elevator adapted to adjust the travel of elevator buckets situated on said elevator, in relation to the bottom of the barge; an inclined adjustable chute situated near the upper end of the elevator; means for adjusting said chute; a counterweighted pivoted baffle at the top of said chute; a vertical hopper adjustable as to length and horizontally as to position; and means for so adjusting said hopper, substantially as and for the purpose described. Seven claims.



lowering said booms, a trolley on such track carrying a bucket, and a hopper carried by said tower and adapted to receive the material from the bucket.

7. In a hoisting machine, the combination with a scow, of a supporting frame carried thereby, a tower arranged to be moved vertically within said frame, two pairs of tracks mounted in said tower, a pair of booms hinged to said tower and having tracks which form continuations of the tracks carried by said tower, a trolley arranged to operate upon one pair of tracks, means for operating the same, a hopper cage arranged to run upon said other pair of tracks, said trolley being arranged to travel within said hopper cage, a pair of hoppers within said cage, and weighing scales supporting each hopper. Eight claims.

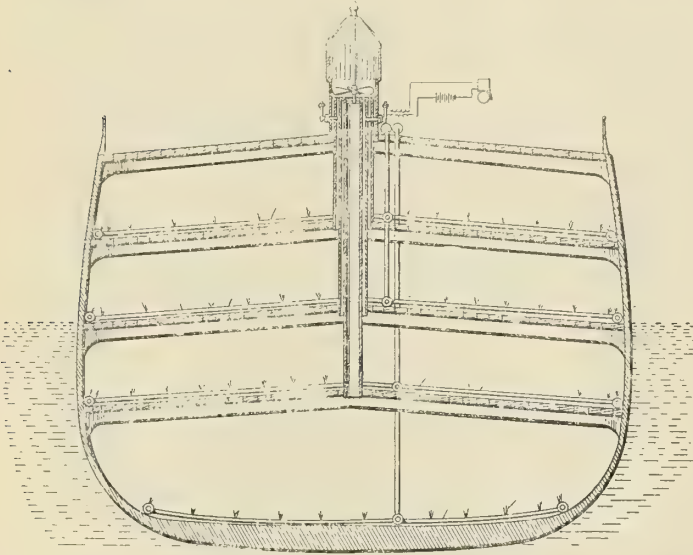


778,702. CURRENT MOTOR. WM. NIEMEYER, ST. JOSEPH, MO.

Claim.—1. A current motor comprising in combination, a water wheel, a shaft on which said wheel is mounted, a gear wheel carried by said shaft, a second gear wheel with which said first gear wheel meshes, a third gear wheel with which said second gear wheel meshes, a shaft carrying the same, means for transmitting rotation from said last shaft, a rotatable bearing for said first shaft at substantially the axis of said second gear wheel, and means for raising and lowering said first shaft about the axis of said bearing. Eight claims.

779,157. MEANS FOR EXTINGUISHING FIRES ON SHIPS. HUGO GRONWALD, BERLIN, GERMANY.

The invention is carried into practice by providing each hold of the vessel with a pipe system for introducing carbonic acid, the said system being preferably so arranged that the carbonic acid or extinguishing gas will rise from the floor of the hold or space, in combination with exit pipes having substantially the same discharge capacity or whose areas in cross section are substantially the same as those of the aggregate carbonic-acid gas inlets to



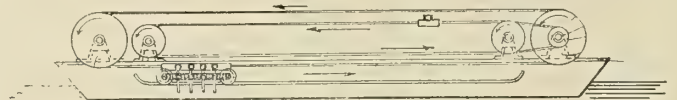
the respective compartments or holds of the vessel. Such pipes are preferably provided at the upper ends with safety valves which are adapted to be properly adjusted. The exit pipe is to serve also to let off the smoke and hot air arising in the case of a fire only to such an extent as is necessary to prevent injurious pressure, but not in such a measure as to prevent the formation of a proper mixture of air and carbonic acid. These safety escape or exit tubes preferably extend to the upper deck and are there provided with suitable thermometers and alarm devices, so that said pipes will serve also for announcing the existence of the fire and facilitating the proper observation of the condition of the same, while the extinguishing operation is proceeding from the deck. Twelve claims.

779,493. LOADING OR UNLOADING MACHINE. JOHN McMYLER, CLEVELAND, O.

Claim.—2. In a hoisting machine, the combination of a supporting frame, a tower carried thereby and having a suitable track, means for raising and lowering the tower, a pair of booms pivoted to the sides of the tower and adapted to form a continuation of the track thereon, means for raising and

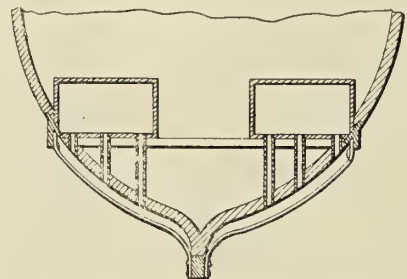
779,600. BILGE BLOCK. JOHN HICKLER, PASADENA, CAL.

Claim.—1. A bilge block comprising a base or supporting element, a tilting member counterweighted at one end and having a bearing tube secured thereto adapted to rock within the bearing surface of the supporting block, said tube having curved flanges upon its outer ends to permit a slight transverse rocking movement to compensate for irregular surfaces, substantially as described. Three claims.



780,071. BOAT AND MEANS FOR PROPELLING SAME. HERMANN STASNY, FRANKFORT-ON-THAINE, GERMANY.

Abstract.—The invention relates to improvements in boats and means for propelling same, said means comprising a slide provided with movable paddles and adapted to be reciprocated longitudinally of the boat and mechanism for operating said slide, the latter being connected to an endless chain driven by a second endless chain adapted to be moved to and fro by means of a handle. Two claims.



780,122. AIR APPARATUS FOR VESSELS. HANS NELSON, CHIPPEWA FALLS, WIS.

The object of the invention is to improve the construction of the hulls of vessels and to provide means for increasing the speed of the latter by uniformly distributing a current of air, gas or similar fluid between the hull and the water in which the hull rests, thereby reducing the frictional resistance of the vessel and materially assisting the driving mechanism in propelling the boat; also to provide novel means for compressing air and forcing the same under pressure longitudinally of the hull and in a direction opposite to the direction of travel of the vessel. Four claims.